WADC TECHNICAL REPORT 52-153

DEVELOPMENT OF FIXED TWO-WAY RESTRICTOR VALVES FOR USE ON MILITARY AIRCRAFT

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WRIGHT AIR DEVELOPMENT CENTER

NOTICES

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DEVELOPMENT OF FIXED TWO-WAY RESTRICTOR VALVES FOR USE ON MILITARY AIRCRAFT

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Aircraft Laboratory

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E. O. No. R452-495

Wright Air Development Center
Air Research and Development Command
United States Air Force
Wright-Patterson Air Force Base, Ohio

McGregor & Werner, Inc., Wakefield, Mass.
Sept. 11, 1952 100

FOREWORD

This report was completed under research and development Expenditure Order No. R452-495 Hydraulic and Pneumatic System and Component Development. It was administered under the direction of the Aircraft Laboratory, Aeronautical Division, Wright Air Development Center, Air Research and Development Command, Lt R. L. Vick and Mr. H. O. Hendrickson acting as project engineers. This is the completed report on this project.

The test articles were purchased from Purolator Products, Inc., under Purchase Order No. (33-038)49-5051-E.

ABSTRACT

Standardization of a fixed two-way 3000 psi restrictor valve, which would be reasonably independent of fluid viscosity changes, was the main object of the development. All tests were conducted at the Wright Air Development Center by the Hydraulics and Pneumatics Section of the Mechanical Branch of the Aircraft Laboratory. The tests and development resulted in an orifice type restrictor valve with protective filters for .067 inch diameter orifices and smaller and a restrictor valve without filters for orifices above .067 diameter. A range of interchangeable standard orifice diameters is a feature of the new valve design. For optimum standardization, it is recommended that one type assembly be used, with unions or adaptors as applicable, for the small orifices needing filters and a larger size type for the larger orifices, not requiring filters.

PUBLICATION REVIEW

Manuscript copy of this report has been reviewed and found satisfactory for publication.

FOR THE COMMANDING GENERAL:

John Gramer, Capt., USAF

for JACK A. GIBBS
Colonel, USAF
Chief, Aircraft Laboratory
Aeronautics Division

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INTRODUCTION

Present-day restrictor valve designs do not have the features necessary for proper performance under all conditions and, in addition, are not standardized for Military use. As a result, and at the request of the Aircraft Industry, the Air Force agreed to undertake this development project in order to develop a valve design which would incorporate the necessary requirements and which could be a standard item for Military procurement.

The Aircraft Industry, acting through the SAE A-6 committee, submitted a tentative range of body and orifice sizes on which to initiate development. The Wright Air Development Center established requirements in conformance with AN standards and added requirements considered necessary for a satisfactory restrictor valve design.

The following features were required of this restrictor valve:

1. Must be designed for 3000 psi operation.
2. Low sensitivity to fluid viscosity changes.
3. Light weight.
4. End bosses to provide maximum adaptability to AN standard fittings.
5. Reliability in service.

Since the simple orifice type restrictor valve incorporates all of the above features except possibly the low sensitivity to fluid viscosity changes, it was decided to investigate the possibility of designing an orifice that would have approximately the same coefficient of discharge at all temperatures.

If the orifice type proves satisfactory, two more features are required:

1. Orifice protection from dirt, etc., in hydraulic fluid.
2. A range of interchangeable orifice diameters.

A contract was awarded to Purolator Products, Inc., to manufacture valve bodies, protective filters, and interchangeable orifice plugs.

A Military Specification and Drawing will be prepared as a result of work accomplished on this project and recommendations of the Aircraft Industry.

SECTION I

DETERMINATION OF THE BASIC RESTRICTOR VALVE ORIFICE DESIGN

As explained in the Introduction, simple orifice type restrictor valves would seem very satisfactory for standardization if an orifice could be designed to have low sensitivity to fluid viscosity changes.

Present orifice type restrictors have varying degrees of wall thickness at the orifice and, through theory and experiments, the sharp edged type has been found to function most consistently at all viscosities. Long orifices are affected considerably by viscosity change especially at high viscosities.

A thin wall (.031 inch length) orifice was designed to give satisfactory performance throughout the temperature range required (-65°F to +160°F).

Tests were run to determine the orifice configuration that would be least susceptible to clogging by particles of O-ring or dirt, etc. The sharp edged orifice again was superior in that a .067 inch diameter sharp edged orifice would pass measured O-ring particles which would not pass through a round edged .095 inch diameter orifice. Extensive tests were performed on this particular phase where various sizes and shapes of O-ring particles were passed through orifices. The results merely emphasized the need of filters for orifice protection, especially in the smaller sizes.

A range of orifice diameters submitted by the Aircraft Industry included the following: .016, .022, .028, .033, .040, .046, .052, .059, .067, .081, .095, .109, and .120 inch. The .016 inch diameter orifice was not flow tested but its pressure drop vs. flow curve could be easily calculated from other data included in this report. This orifice would be more sensitive to viscosity changes than the larger diameter orifices.

Previous service experience with orifice type restrictor valves indicated that all orifices below .070 inches in diameter should be protected by filters. In the clogging tests it was found that in sizes above .067 inch orifice diameter, the probability of complete clogging was remote insofar as O-ring particles were concerned. If O-ring particles exist in a system in such large sizes that they clog the larger orifices, filter protection for other hydraulic components having small openings would be required and this has not been found necessary. Thus, filters are needed for orifices of .067 inch diameter and smaller.

SECTION II

DEVELOPMENT OF AN ORIFICE PROTECTIVE FILTER DESIGN

Orifice filter element tests were conducted first since the results of these tests would affect the entire design.

The first filter elements tested, were constructed by winding stainless steel wire in a tubular form with the adjacent turns separated by .003, .005, and .008 inch spaces. They were similar to those shown in Plate 2 except that the pointed end shown was then a flat plate with drilled holes for fluid passage, and the circumference of the plate was threaded so the filter screwed down into the body.

Initial pressure drop tests at -65°F, with MIL-0-5606 fluid containing one percent water by volume, showed that the .003 and .005 sizes would clog during flow conditions. Examination under a microscope revealed small particles of gelatin-like material. New designs were submitted with .008 and .012 inch spacing between the turns. Cold temperature conditions of the fluid would not clog either of these filter elements.

The filter elements were then tested to determine their ability to withstand pressures and flows. Preliminary tests showed that the high forces induced by the jet action of the fluid leaving the orifice, stretched the filter elements. Another force, perpendicular to the first force, caused by the jet striking the end of the filter elements, caused the elements to either expand or to break. The combination of these forces caused nearly all of the filters to fail. The .095 inch orifice with 3000 psi pressure drop appeared to produce the largest jet forces and this size was used to determine strength adequacy if no failure occurred with smaller orifices. Smaller orifices, which produced smaller forces, did not cause the filters to fail. The failures occurred at the welds between the individual coils. Tests were conducted where the stretching force was eliminated and the elements withstood the jet action until extremely high flows were reached.

The -6 size, .012 spaced filter failed after three minutes at 2500 psi and 9.2 gpm (gallons per minute). The -6 size, .008 spaced filter failed after four minutes at 3000 psi and 5.1 gpm. The tensile load was eliminated by removing the threads from the circumference of the flat plate on the end, thus allowing the element to ride free for a short distance along the axis of the restrictor body. The tensile load was caused by not turning the element down tight against the seat in the body thus allowing it to stretch until it did reach the seat. Such a condition could readily occur in service.

The -8 size, .012 spaced filter failed under 2500 psi and 9.45 gpm with the tensile load present, and at 2500 psi and 9.5 gpm after five minutes with the tensile load eliminated. The .008 spaced filter, with the tensile load eliminated, failed at 2500 psi and 9.5 gpm after four minutes.

The -10 size, .012 spaced filter failed after being subjected to 2500 psi and 14.5 gpm for three minutes, with the tensile load present, and after being subjected to 3000 psi and 15.9 gpm for one minute, with the tensile load eliminated. The .008 spaced filter failed after three minutes at 3000 psi and 15.8 gpm, with the tensile load eliminated.

All of the failures occurred at greater than line size flow and were due, as mentioned above, to the jet action of the flow from the orifices.

Present experience indicates that any size body with filters should not fail regardless of the size of the orifice used. Inasmuch as the jet forces were not considered a factor in the original filter design, two new types, designed to withstand these jet forces, were submitted. One was the same brazed steel type as previously tested except that a pointed breaker was attached at the end nearest the orifice to break the jet action of the fluid and the threads were removed to eliminate the tensile load. The other type consisted of stainless steel wire, with rises, wrapped around pinion stock to form filter spaces and a pointed end which was slightly different from the brazed type end as shown in Plates 1 and 2.

The -8 and -10 sizes in both types were tested with the .095 orifice at 3000 psi without failure.

The -4 and -6 sizes were tested and showed the stainless steel type to be acceptable. The brazed type -6 size collapsed from end loads and the -4 size broke at the middle in a manner similar to earlier failures. The Purolator Company revised these designs to incorporate a rod in the center to take compression loads. Flow tests were conducted at 3000 psi for five minutes without failure.

The centers were partially drilled out of the stainless steel pinion stock type but they are still comparatively heavy.

It was not considered necessary that the filter spacings be appreciably smaller than the smallest orifice diameter and in order to reduce the possibility of clogging and possibly to reduce sensitivity to fluid viscosity changes, the .008 and .012 inch spacings were considered to be the most advantageous.

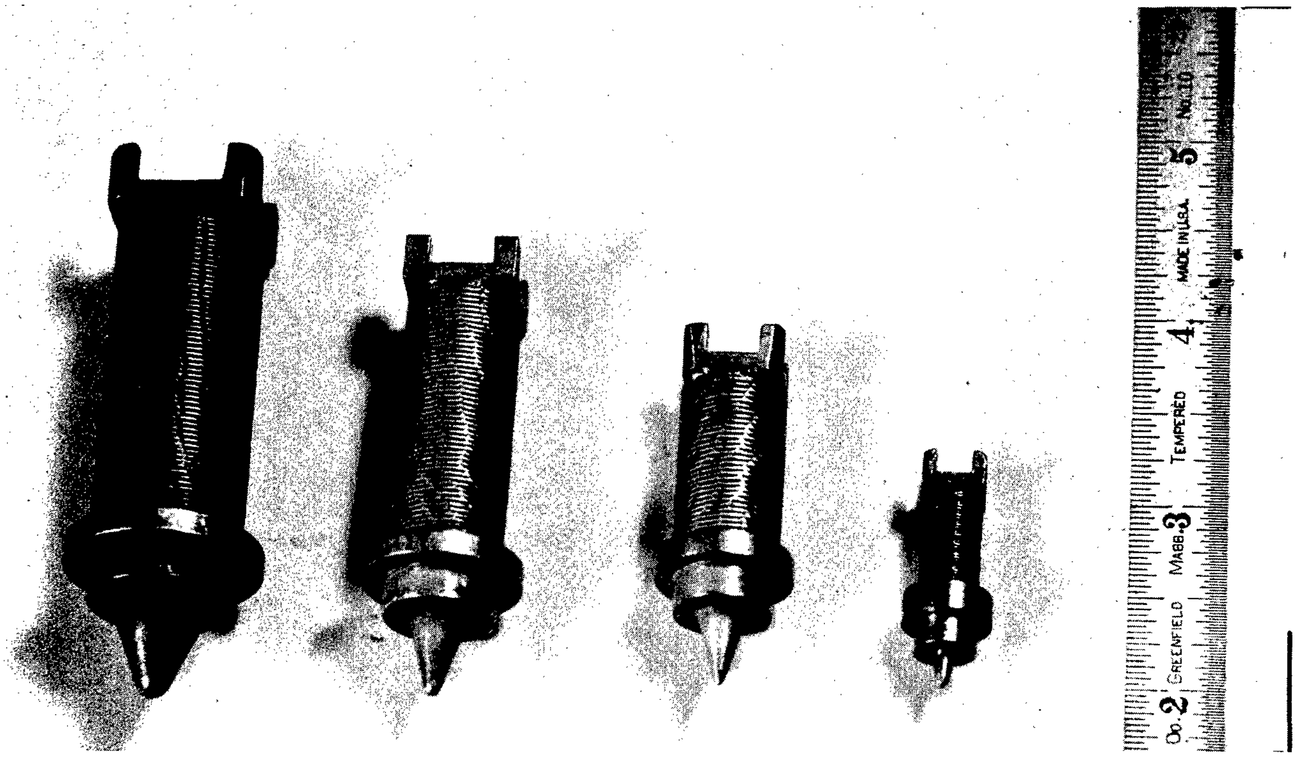


PLATE 1

STAINLESS STEEL PINION
STOCK TEST FILTERS -10, -8,
-6, AND -4 SIZES

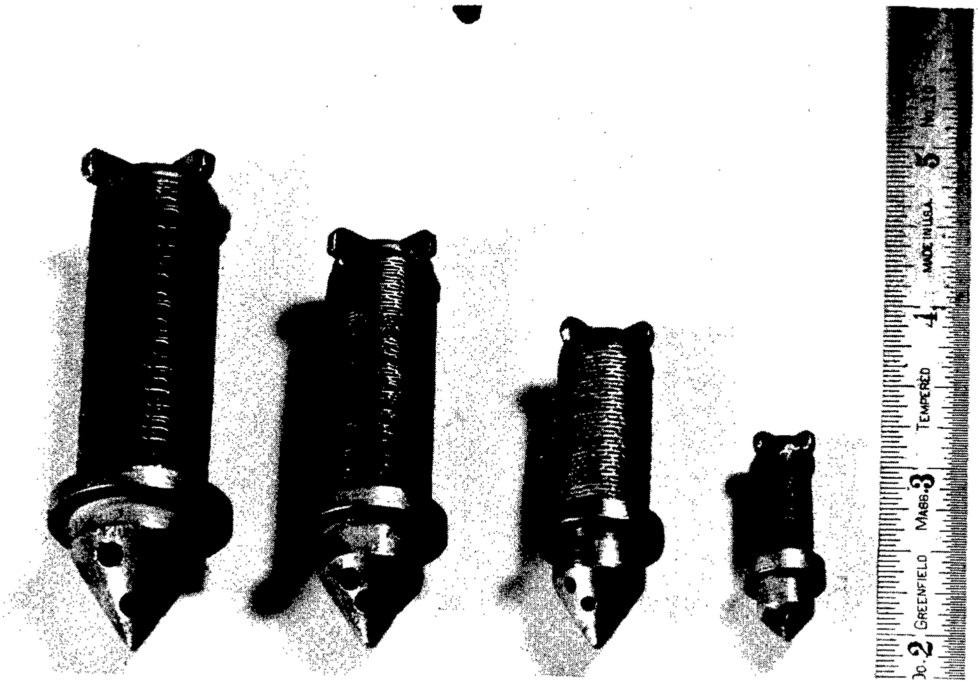


PLATE 2

BRAZED STAINLESS STEEL
TEST FILTERS -10, -8, -6,
-4 SIZES

The .008 and .012 inch spaced brazed type rod reinforced filters were considered to be the best all around.

Pressure drop tests at various flows were conducted and are explained later in the report.

SECTION III

TESTS ON ORIFICES AND ASSEMBLIES AND THEORY ON PRESSURE DROP AND FLOW

General

Wear and strength tests were required to determine whether or not the .031 inch thickness at the orifice was enough so that the flow characteristics would not be changed throughout the life of the valve. These tests were conducted before the pressure drop and flow tests since the pressure drop and flow tests would require considerable work and any change in design should be accomplished prior to such extensive tests.

Wear Test

A flow test to determine wear was conducted at room temperature on an .067 inch diameter orifice with 3000 psi pressure drop across the orifice at 5.2 gpm flow for 10 hours. No significant wear was noted. Measurements of the orifice diameter with a measuring microscope, before and after the test, showed an increase of approximately .0005 inches across the diameter in one direction. At 90 degrees to this direction, measurements across the diameter indicated no change. A slight erosion was also noted. The 10 hour test on the valve was considered equivalent to a life wear test and it was concluded that wear or erosion would not be a problem. See Figure 1B.

Impulse Test

Impulse tests were conducted at room temperature on the -6 size proposed design assembly. The valve, with an .016 orifice and two type SK 18118 rod reinforced filters installed and one end capped, was subjected to 50,000 impulse cycles from 0 to 3000 psi pressure with 4500 psi peaks. There were no failures during the test. The same valve assembly, except for a .120 orifice installed in place of the .016 orifice, was tested for 50,000 additional cycles without evidence of permanent distortion or failure. See Figure 1A.

Endurance Test

The same body and filters, as used in the impulse test, were endurance cycled with an .067 inch orifice for 20,000 flow reversals. The pressure cycle was from 0 to 3000 psi with no peaks. The assembly did not fail or permanently distort in any way. See Figure 1B. The fluid was cycled slowly to prevent extreme heating. The above test results were considered satisfactory.

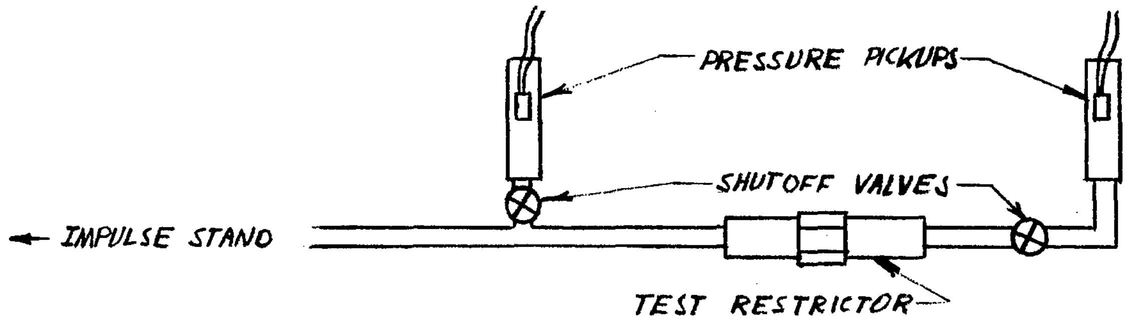


FIGURE 1A - IMPULSE TEST DIAGRAM

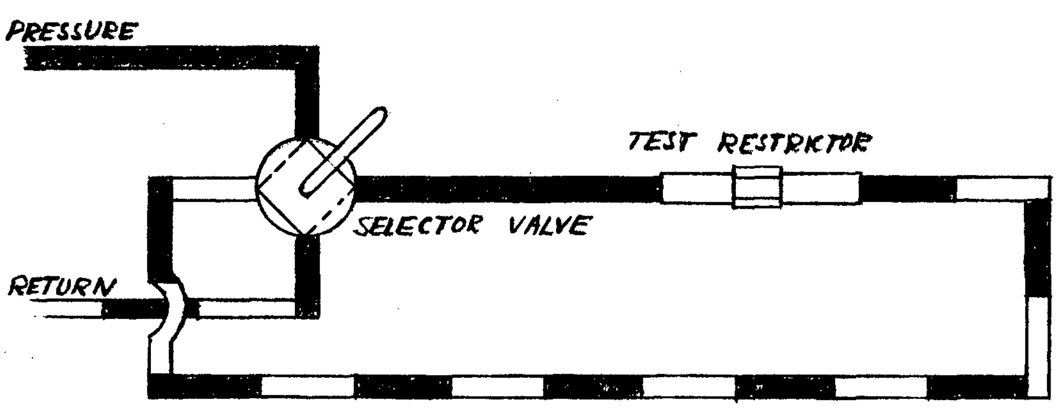


FIGURE 1B - WEAR AND ENDURANCE TESTS

Pressure Drop Through Orifices at Room Temperature

To determine the pressure drop at various flows through the orifices at room temperature, the test was set up as in Figure 2A for reasons explained later in this section.

On this setup, the restrictor flowmeter was calibrated by using pressure pickups (explained later). The selector valve was placed in neutral, the shutoff valve to the flowmeter opened and flow controlled by the bypass valve on the test stand. The flow was read on the stand flowmeter and plotted against the deflection on an oscillograph caused by the flow through the restrictor flowmeter (explained later.)

The tests were conducted as follows:

1. The portion of the test set-up between the No. 1 transfer cylinder bottomed and the No. 2 on top, or vice versa, was filled with hydraulic fluid conforming to Specification MIL-O-5606. The air was bled from the setup.

2. The shutoff valve from the pressure line to the restrictor flowmeter was then closed.

3. The selector valve was placed in the neutral position.

4. The pump was started and various line pressures up to 3100 psi were controlled by using the bypass valve. (Test stand line pressure gage not shown).

5. The selector valve was placed in the position shown when the desired pressure was reached. The fluid pushed the piston in the No. 1 transfer cylinder up, causing the fluid in the upper portion of the cylinder to be forced through the test restrictor and in turn forcing the piston in No. 2 transfer cylinder down, which forced the fluid below the piston in the No. 2 cylinder back through the selector valve and through the restrictor flowmeter. The pressure pickups at the test restrictor had previously been calibrated so that the pressure drop across the test orifice could be obtained, along with the flow, as it passed through the calibrated restrictor flowmeter. The actual flow and recording on the oscillograph could take place in approximately 4 seconds.

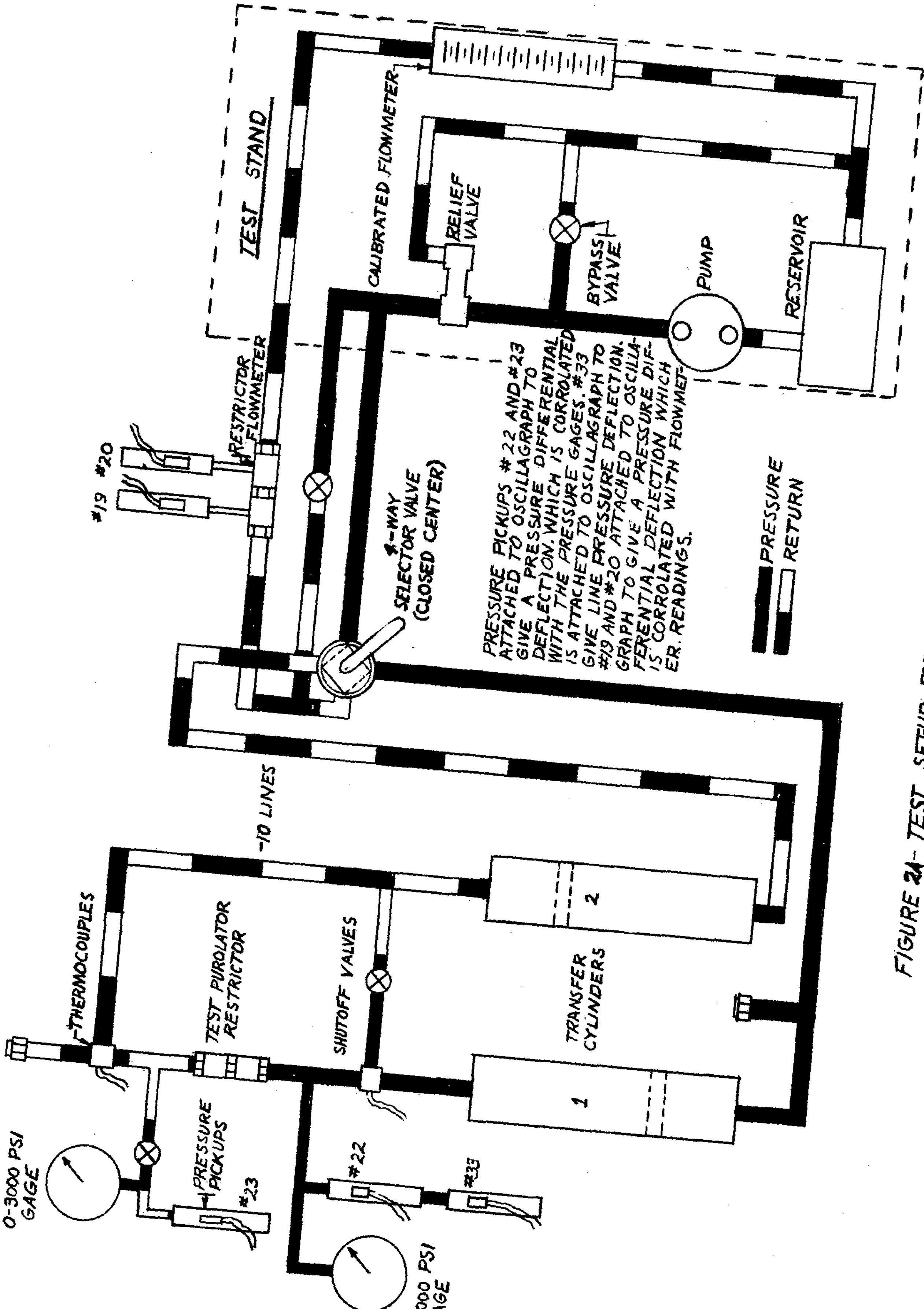


FIGURE 2A- TEST SETUP FOR THE RESTRICTOR VALVE

Previous experience with testing orifices in the cold room indicated that the only practical method of measuring flow and pressure drop simultaneously was by the use of instrumentation rather than by visual observation of a flowmeter and bourdon tube type pressure gages. Because of the heating of the fluid during flow, the duration of flow had to be kept short and accurate visual reading of the instruments was not possible. The instrumentaion used, consisted of a 6 channel Miller recording oscillograph, a Miller CD-2 amplifier, strain gage type pressure pickups, and thermocouples. The pickups consisted of 1 inch OD x .120 inch wall thickness 61ST aluminum alloy tubing with reduced wall sections at the point of strain gage application. A two channel bridge was constructed to permit two pressure pickups to form a four arm bridge and wired so that, with equal pressure on the pickups, the bridge would remain in balance. An adjustment was provided to permit equalizing the sensitivities of the pickups. The output of the bridge was fed directly to 35 cps, 5 micro amp/in. galvanometers in the oscillograph. This arrangement permitted the recording of pressure drop or flow with one galvanometer trace. One channel was calibrated for measuring flow in the return line and another for pressure drop across the test restrictor valve. Line pressure was recorded by use of the amplifier. Thermocouples on either side of the test valve and attached directly to 35 cps galvanometers gave a record of the fluid temperatures during flow conditions.

The traces were calibrated by plotting graphs of galvanometer deflection against flow rate or pressure drop as applicable. From these graphs (not included) the flow rates and pressure drops were found and plotted.

Symbols Used in Theoretical Analysis

Symbol	Description	Unit
a	Area	Square Inches
C	Coefficient of discharge	No units
d	Diameter of orifice	Inches
d_1	Diameter of tube in which orifice is placed	Inches
KN_R	Reynolds Number times constant K, which represents the equivalent diameter of the particular article being analyzed. KN_R changes for every test article.	No units
K_1C	Coefficient of discharge times constant K_1, which represents the equivalent area of a particular article being analyzed. K_1C changes for every test article.	No units
N_R	Reynolds Number	No units
PD	Pressure drop	Pounds per square inch
Q	Flow	Gallons per minute
γ	Kinetic viscosity	Centistokes
γ_t	Kinetic viscosity due to temperature.	Centistokes
γ_{pf}	Pressure factor on kinetic viscosity. Multiply γ_t times γ_{PF} to get γ.	No units

Ω, ϕ, Ψ, N, g, P_1, P_2, W_1 See Reference 1 (Bibliography). Q in Reference 1 (Bibliography) is in different units than that listed above.

Some Theory on Orifice Flow

The test orifices were used with -10 bodies since the pressure drop through these bodies is negligible at room temperature.

The coefficient of discharge C for similar orifices, is a function of Reynolds Number. Actually the magnitude of the ratio of the diameter of the orifice to the diameter of the tube in which it is placed (d/d_1) and also the absolute size, as distinguished from relative size or geometric similarity, affect the value of C slightly.

The maximum (d/d_1) ratio in the test setup was less than 0.2. In Reference 1 (Bibliography), Q is a function of $(\Omega\phi)$ $N\sqrt{2g(P_1-P_2)W_1}$ where $\sqrt{2g(P_1-P_2)W_1}$ is the head loss. From a (d/d_1) range of 0 to 0.2, the coefficient $(\Omega\phi)$ varies only slightly and the variation can be considered negligible for normal room temperature. N changes directly with the area of the orifice and inversly to the square root of $1-(d/d_1)^4$. $N = \frac{a}{\sqrt{1-(d/d_1)^4}}$. For $(d/d_1) = 0.2$ or less, the square root term is negligible.

According to Reference 3 (Bibliography), the effect of absolute size on C is generally expressed as a function of the surface grain size with respect to the pipe diameter d_1, or as a function of the relative roughness of the pipe section preceding the orifice. This effect, however, is small and may be either positive or negative, depending on the geometrical construction of the approach.

It is concluded that for normal room temperature and the range and design of the orifices used, the effect of d/d_1 and absolute size on C is negligible and can be neglected.

At low Reynolds Numbers the curve of N_R vs. C changes. John L. Hodgson, the author of Reference 1 (Bibliography), plotted curves of $\sqrt{N_R}$ vs. C for various orifices. Individual $\sqrt{N_R}$ vs. C curves could be plotted for the orifices but with the range of orifice diameters to be considered for standardization, the curves would be almost exactly the same for $\sqrt{N_R} = 60$ and above and, as seen from the plotted curve (Figure 5), the resulting points are erratic; therefore, greater accuracy would be obtained by using one curve for this portion with very slight errors in C as explained. According to Reference 1, the slope of the curve $\sqrt{N_R}$ vs. C for $\sqrt{N_R}$'s less than 1.5 depends on a constant ψ. Analysis of the data given on ψ indicated that for the range of (d/d_1) ratios used in the test, the difference in slope is negligible.

The range of $\sqrt{N_R}$'s between 1.5 and 60 was plotted from test data and followed a definite trend as shown in Figure 5. A curve of N_R vs. C was plotted to include all of the test orifices used. (Figure 5)

The orifices were measured and henceforth all of the test orifices will be designated by their actual size. Sizes are .0221, .0295, .0361, .0407, .0461, .0524, .0586, .0676, .0813, .0965, .1092, and .1196.

Simple equations had to be developed to make calculations. Basic equations for N_R and C are: $N_R = \frac{V(ft/sec)\, d(ft)}{\nu\,(ft^2/sec)}$ and $C = \frac{Q}{A\sqrt{PD}}$.

Flow has been measured in gallons per minute and the graph for kinematic viscosity vs. temperature is in centistokes so N_R can be simplified.

$$N_R = \frac{V\,d}{\nu}$$

$$V(ft/sec) = \frac{Q(cfs)}{A(sq\ ft)}$$

$$Q\,(cfs) = \frac{Q\,(gpm)}{7.48 \times 60}$$

$$V(ft/sec) = \frac{Q\,(gpm)\,144}{7.48 \times 60\ A(sq\ in)} = .321\ \frac{Q\,(gpm)}{A\,(sq\ in)}$$

$$ft^2/sec = K\ \ cm^2/sec = 1.076 \times 10^{-3}\ cm^2/sec$$

$$ft^2/sec = 1.076 \times 10^{-5}\ \ centistokes$$

$$N_R = \frac{.321\ Q\ (gpm)\,d(in)}{A(sq\ in)(12)(1.076 \times 10^{-5}\,\nu\,cs)} = 3165\ \frac{Q\ (gpm)}{d\ (in)\,\nu\ cs}$$

$\nu = \nu_t \times \nu_{pf}$ where ν_t is kinematic viscosity from temperature (Figure 3) and ν_{pf} is the pressure factor due to pressure at that temperature (Figure 4).

Reynolds Numbers and coefficients of discharge were calculated and plotted in graphic form on semi-log paper. (Figure 5)

Before the aforementioned method was tried, the normal tempera ture curve of N_R vs. C was obtained as follows:

The form of the pressure drop curves was parabolic with an equation; $Q = k\ PD^n$ where k is a constant and n is an exponent.

Using logs: $\text{Log } Q = k + n \text{ Log } PD$

This method was used for all the room temperature runs to determine k and n for each orifice. It was found that n also plotted a parabolic curve when plotted against the orifice diameter and $n=0.573D^{(.05725)}$. Using these values of n, average values of k were found for each orifice and these values also formed a parabolic curve with the diameters of the orifices and $k=13.25D^{(1.7925)}$, therefore, $Q=13.25D^{(1.7925)}PD^{(.573D)^{.05725}}$.

Curves plotted with this equation were all room temperature curves and corresponded within 50 psi in all cases with curves plotted with the N_R vs. C curve. Some calculated curves gave higher flow rates and some lower than the test results but curves calculated both ways were always very close together. These curves are not included because they were calculated before the stand flowmeter was calibrated, however, the results serve to further prove the use of one curve for all of the test orifices and that this curve should be especially accurate at room temperatures.

Cold Room Testing of the Orifices

Since the determination that an N_R vs. C curve could be plotted, the temperature did not necessarily have to remain at -65°F. The simple setup shown in Figure 2B was devised for these tests and the flow was controlled by changing the angle of the variable volume pump instead of using the bypass valve. In this manner the fluid did not warm up so rapidly. The Revere flowmeter was calibrated and the Lewis temperature potentiometer was checked for accuracy. If the temperature increased to -55°F, the fluid was allowed to cool down before continuance of tests.

The pressure drop was obtained by subtracting the reading on the return side of the test valve from the reading on the pressure side. The flow was read directly from the flowmeter and corrected. Temperature was read from the potentiometer.

Line pressure, back pressure, temperature, and flow were recorded and N_R and C were found and plotted on the low end of the N_R vs. C curve. This was done for all orifices and the points followed a definite trend so the curve of Reynolds Number vs. coefficients of discharge was plotted for the complete temperature range of all the orifices.

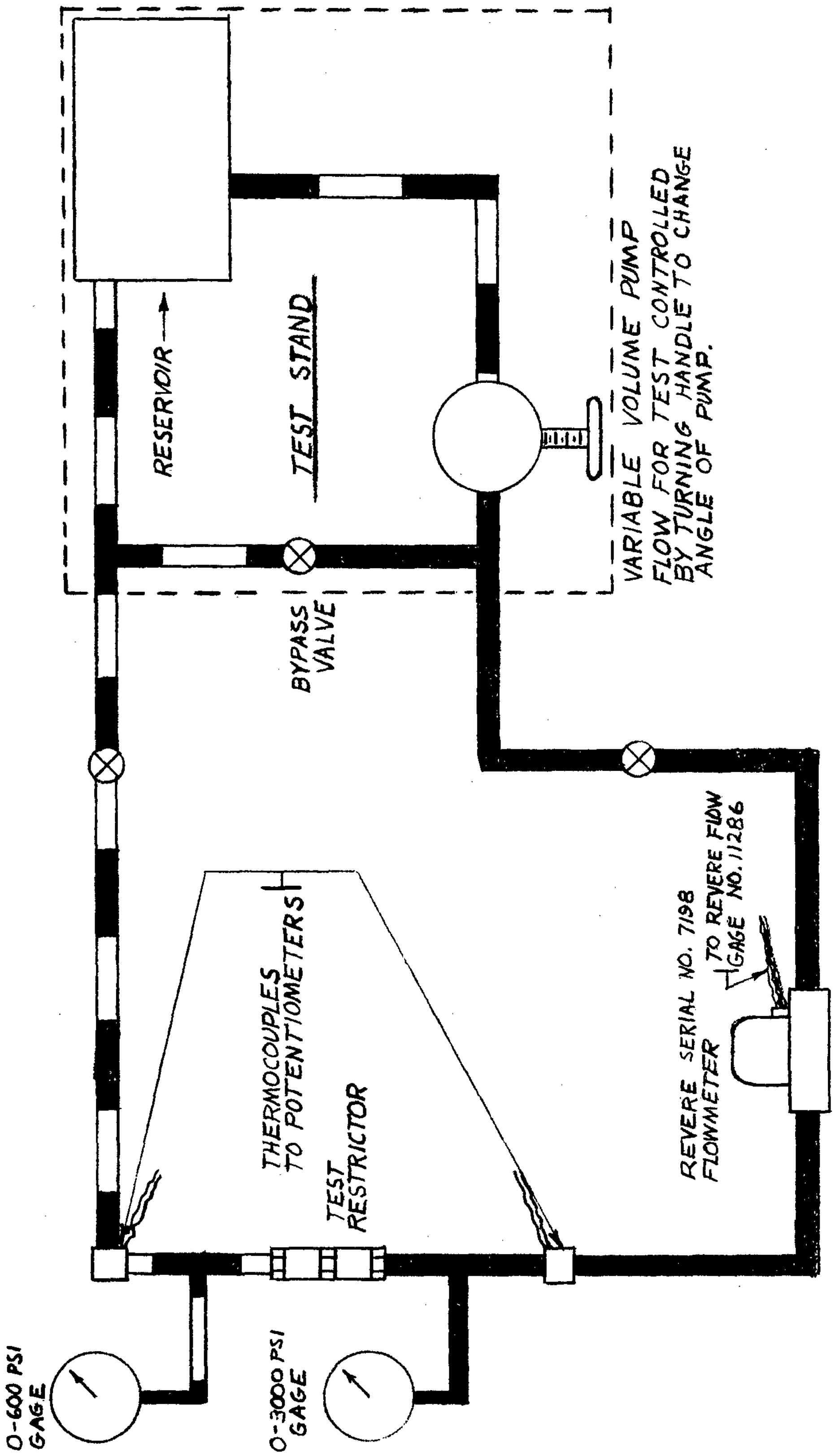


FIGURE 2B-COLD ROOM RESTRICTOR VALVE TEST SETUP

Use of Reynolds Number vs. Coefficient of Discharge Curve

The N_R vs. C curve is shown in Figure 5. It is seen that after a N_R of approximately 1200 the coefficient of discharge becomes constant at 26.75. This means that with a 200 psi pressure drop and MIL-O-5606 at approximately +69°F or above, all of the orifices have the same coefficient of discharge. With a higher pressure drop or using the larger orifices, the constant coefficient of discharge can be at a lower temperature. With the .120 orifice and a 200 psi pressure drop, the coefficient of discharge would be constant for this orifice at approximately +3°F and higher.

For all orifices within the temperature range where C=26.75 the equation for the flow vs. pressure drop curve is $Q=26.75\ A \times \sqrt{PD}$. This equation was used for all of the calculated curves shown with the test curves in Figures 6 and 7.

For the calculated curves at -65°F, the following method was used:

$$N_R = \frac{3165\ Q}{d\ \sqrt{t}\ \sqrt{pf}} \qquad \text{and } Q=CA\sqrt{PD}$$

$$N_R = \frac{3165\ CA\ \sqrt{PD}}{d\ \sqrt{t}\ \sqrt{pf}} = \frac{2485Cd\ \sqrt{PD}}{\sqrt{t}\ \sqrt{pf}}$$

$\sqrt{t}$ at -65° is approximately 1700 centistokes according to Figure 3.

$$N_R = \frac{1.462\ Cd\ \sqrt{PD}}{\sqrt{pf}}$$

Various pressure drops were chosen to plot a full curve. The pressure drops used were 600, 1200, 1800, 2400, and 3000 psi. At 600 psi pressure drop, assuming the pressure after the orifice to be zero, $\sqrt{pf}$ is approximately 1.15 (Figure 4). The pressure before the orifice should be used to determine this factor $\sqrt{pf}$; however, the pressure after the orifice was considered zero for all of the plotted curves, therefore, pressure drop in terms of psi was also considered the pressure of the fluid before the orifice. At 600 psi pressure drop, N_R = 31.15 dC and Q = CA (24.5). The .067 inch orifice gives N_R = 2.085C. With a slide rule and Figure 5, N_R is found to be 58.75 and C is 28.1, which gives a flow of 2.425 gpm. Thus, the flow for 600 psi pressure drop through an .067 inch orifice at -65°F is 2.425 gpm.

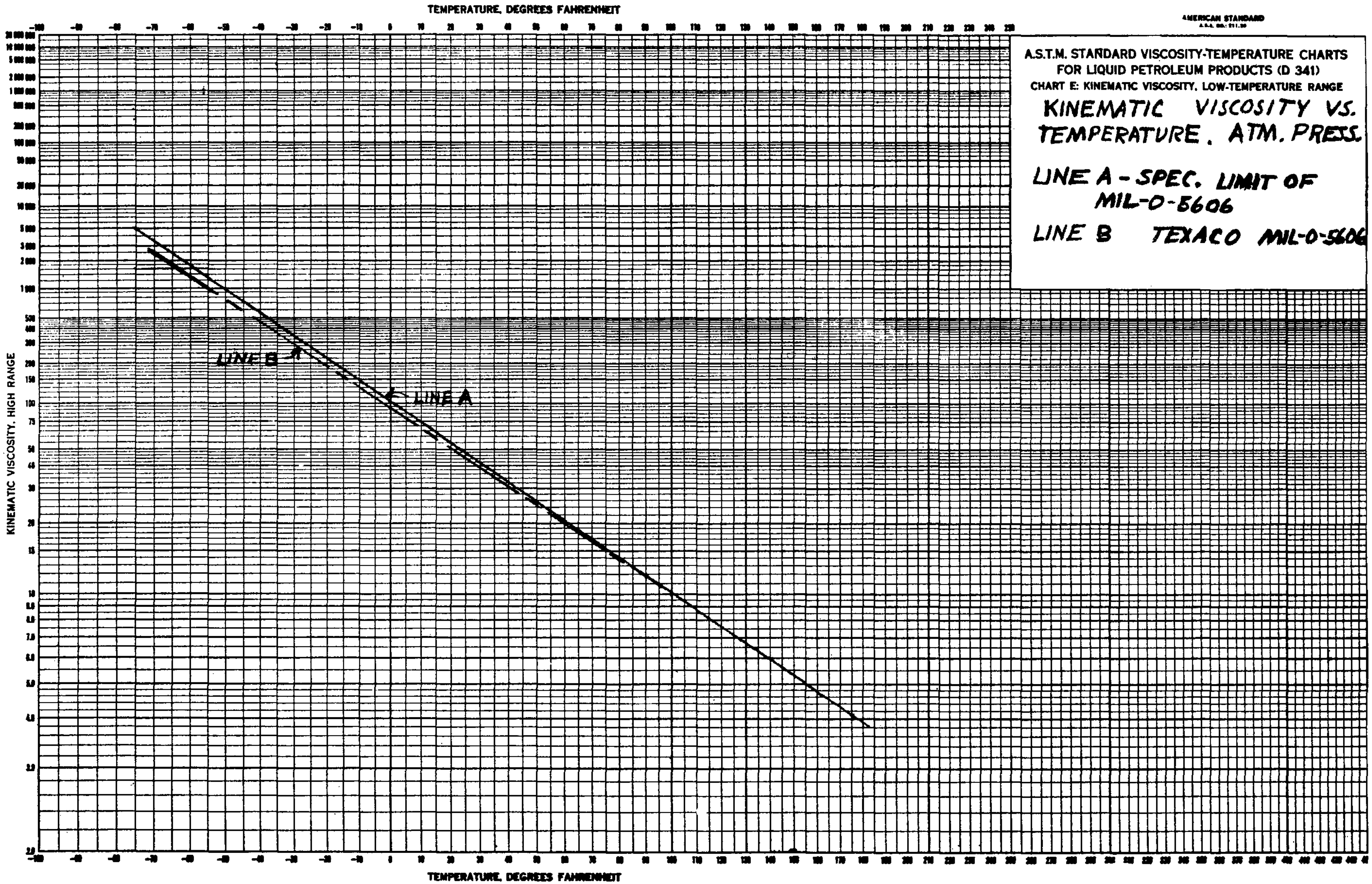


FIGURE 3
KINEMATIC VISCOSITY VS. TEMPERATURE
OF MIL-O-5606 AT ATMOSPHERIC PRESSURE

FIGURE 4

PRESSURE RELATIONSHIP ON KINEMATIC VISCOSITY OF MIL-O-5606 AT VARIOUS TEMPERATURES. REFERENCE NO. 4 PAGE 12 (BIBLIOGRAPHY)

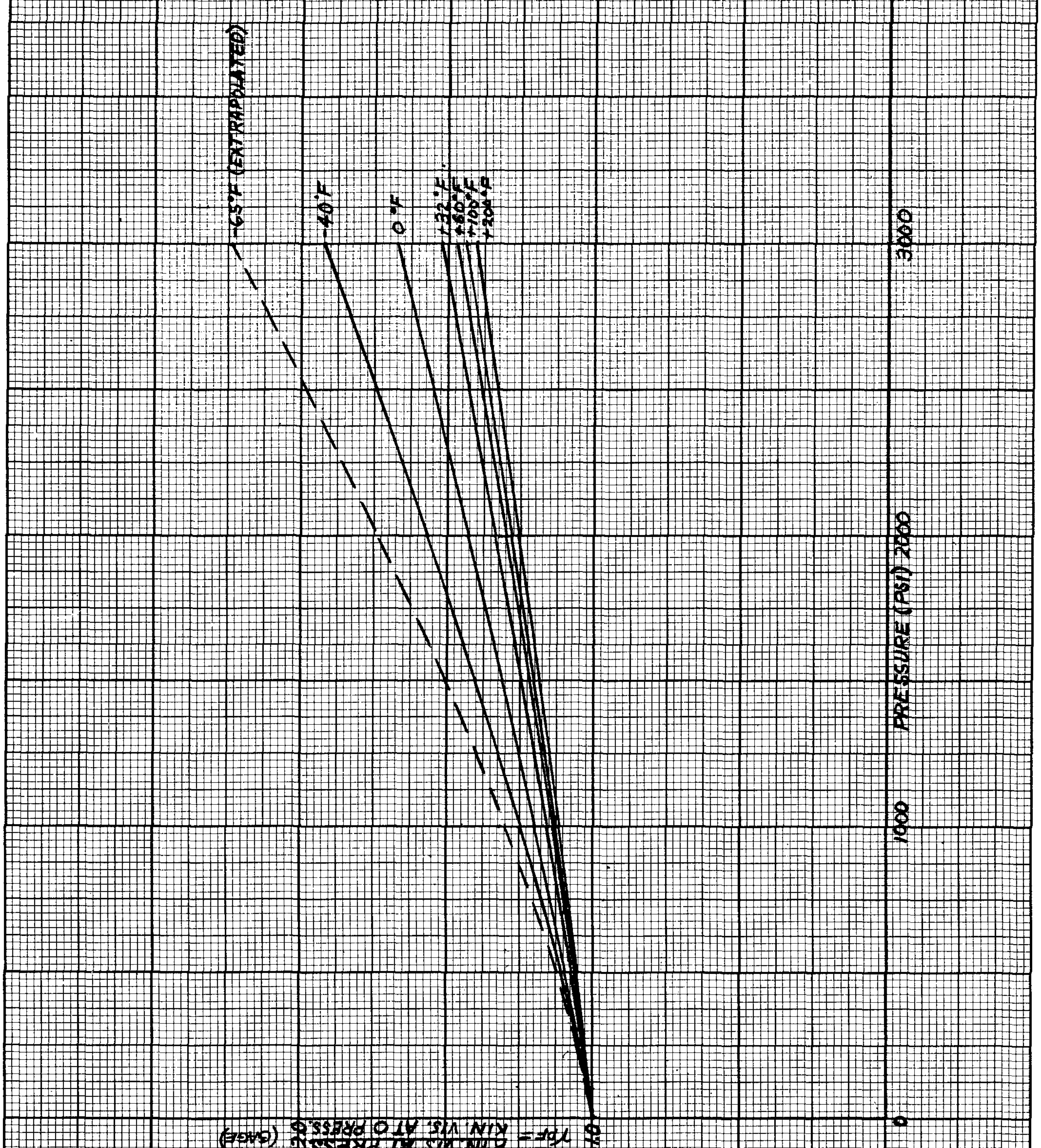
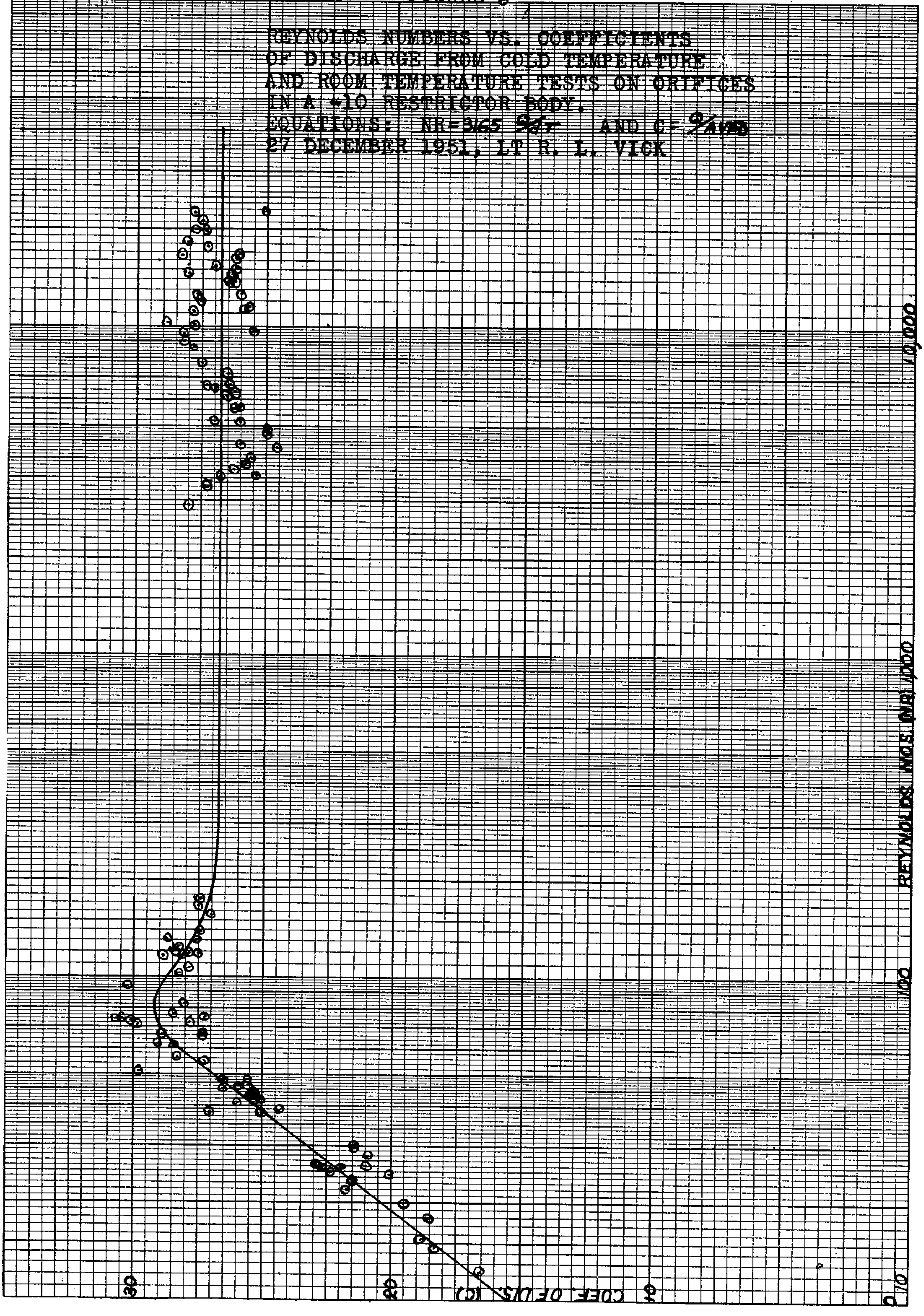


FIGURE 5



REYNOLDS NUMBERS VS. COEFFICIENTS
OF DISCHARGE FROM COLD TEMPERATURE
AND ROOM TEMPERATURE TESTS ON ORIFICES
IN A #10 RESTRICTOR BODY.
EQUATIONS: NR=3165 Q/dv AND C=Q/A√ΔP
27 DECEMBER 1951, LT R. L. VICK

FIGURE 6

PRESSURE DROP VS. FLOW FOR .0221, .0295, .0361, .0407, AND .0461 ORIFICES
MIL-O-5606 FLUID. 28 DEC 51 LT R. VICK

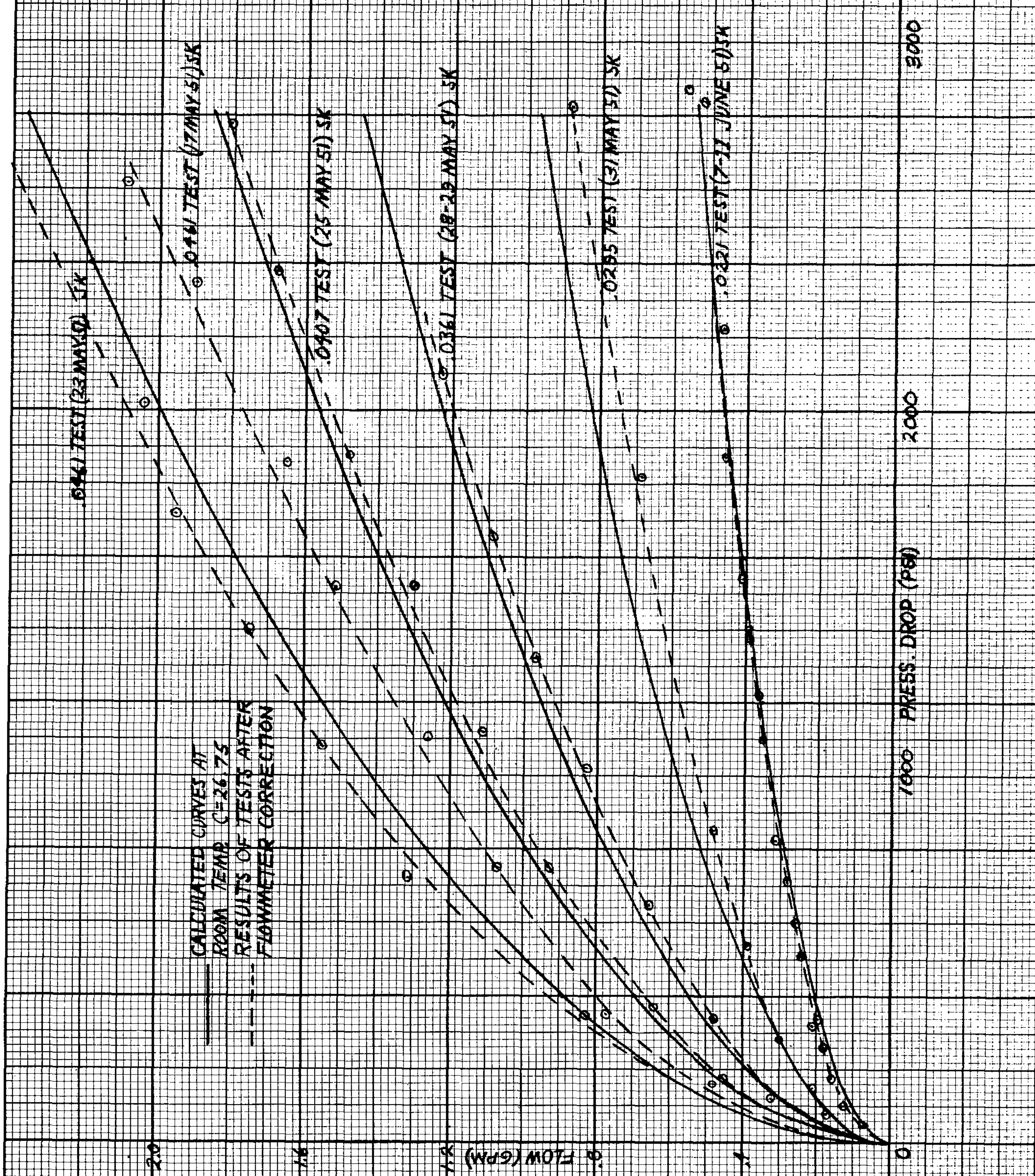


FIGURE 7

PRESSURE DROP VS. FLOW FOR .0524, .0586,
.0676, .0813, .0965, .1092, AND .1196
ORIFICES, MIL-O-5606 FLUID, 28 DEC 51
LT R. L. VICK

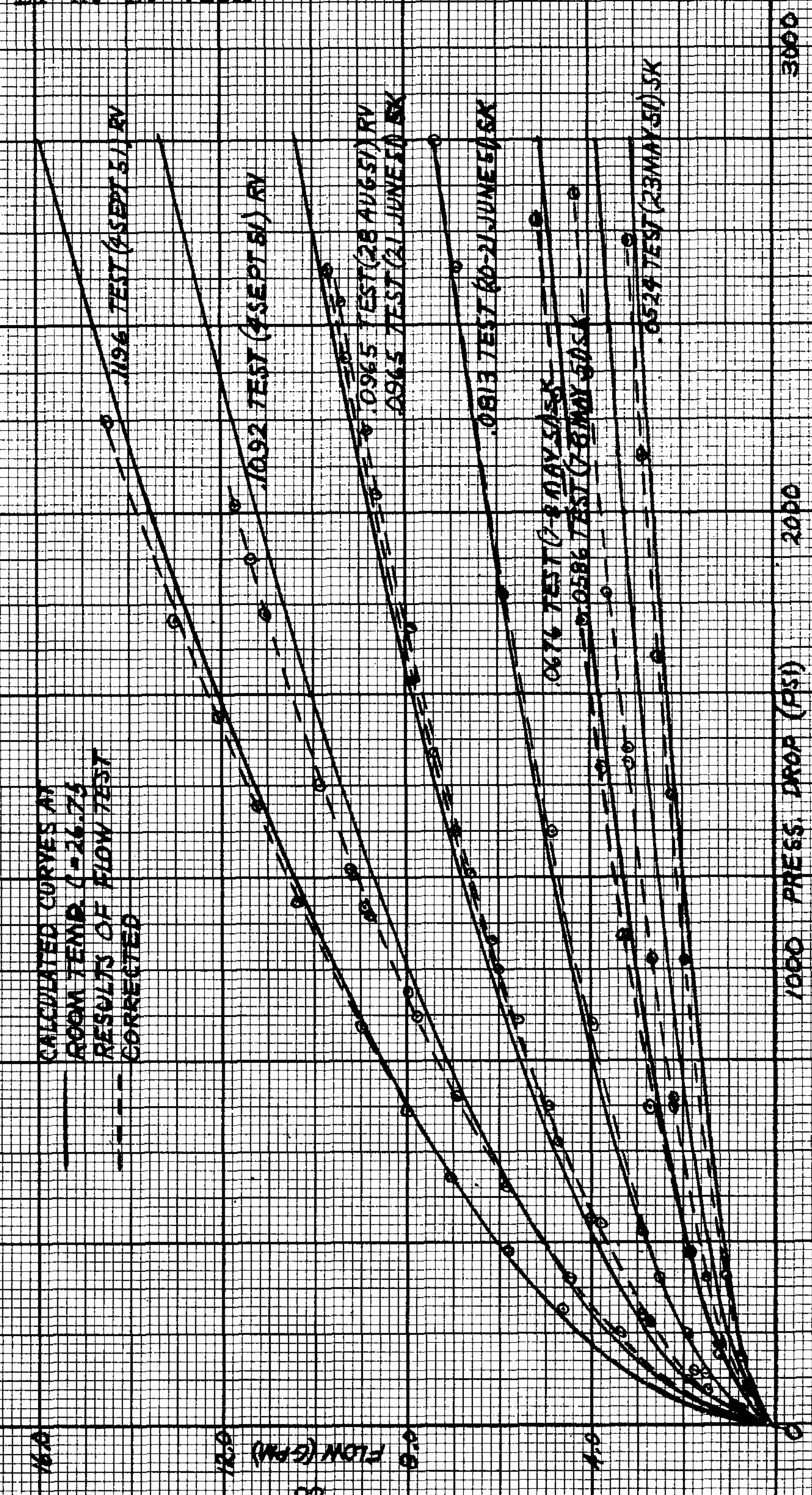


FIGURE 8

COLD TEMPERATURE PRESSURE DROP VS.
FLOW TEST RESULTS ON .0221, .0295,
.0361, .0407, AND .0461 TEST ORIFICES
29 DECEMBER 1951 LT R. VICK

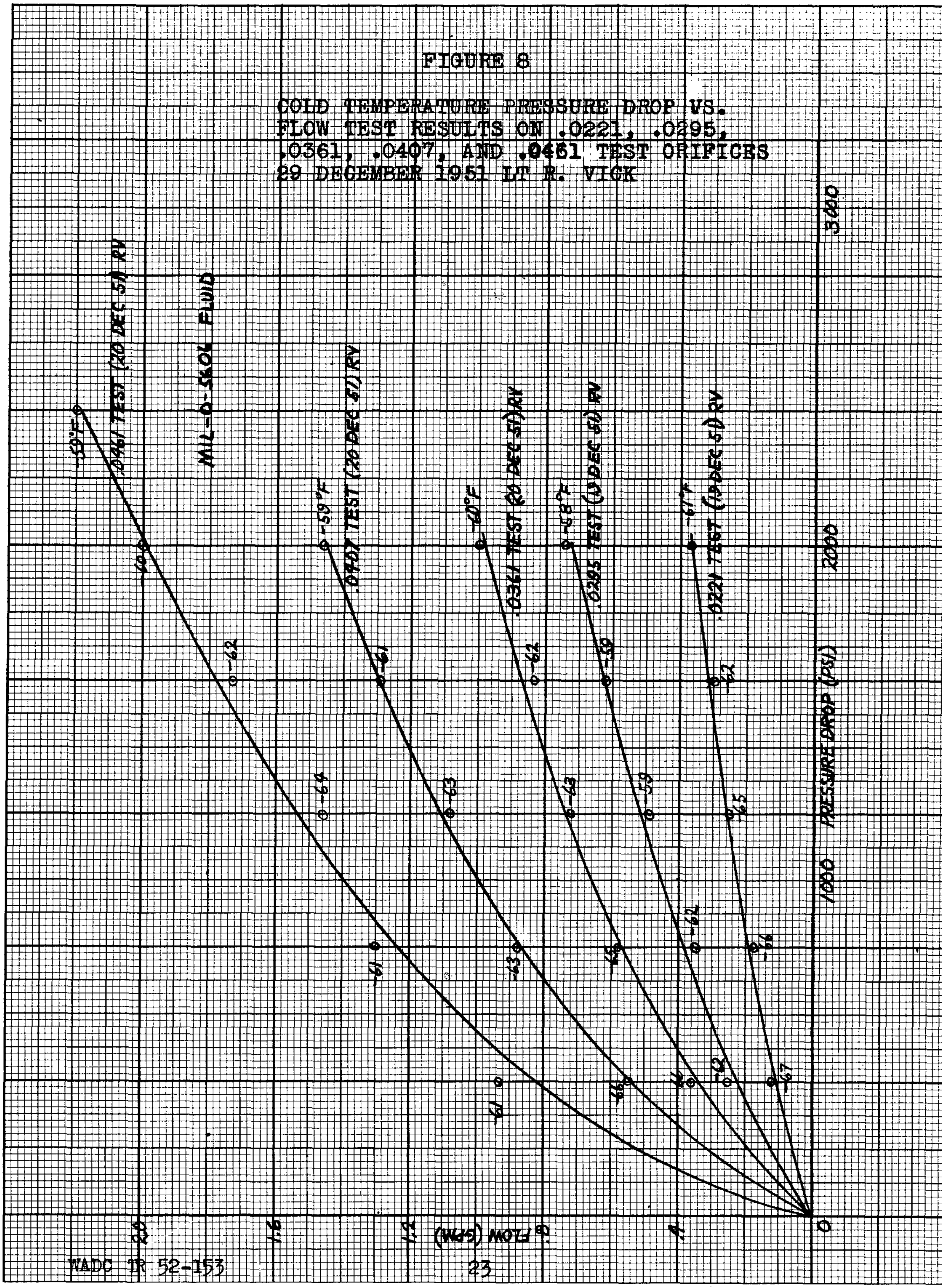
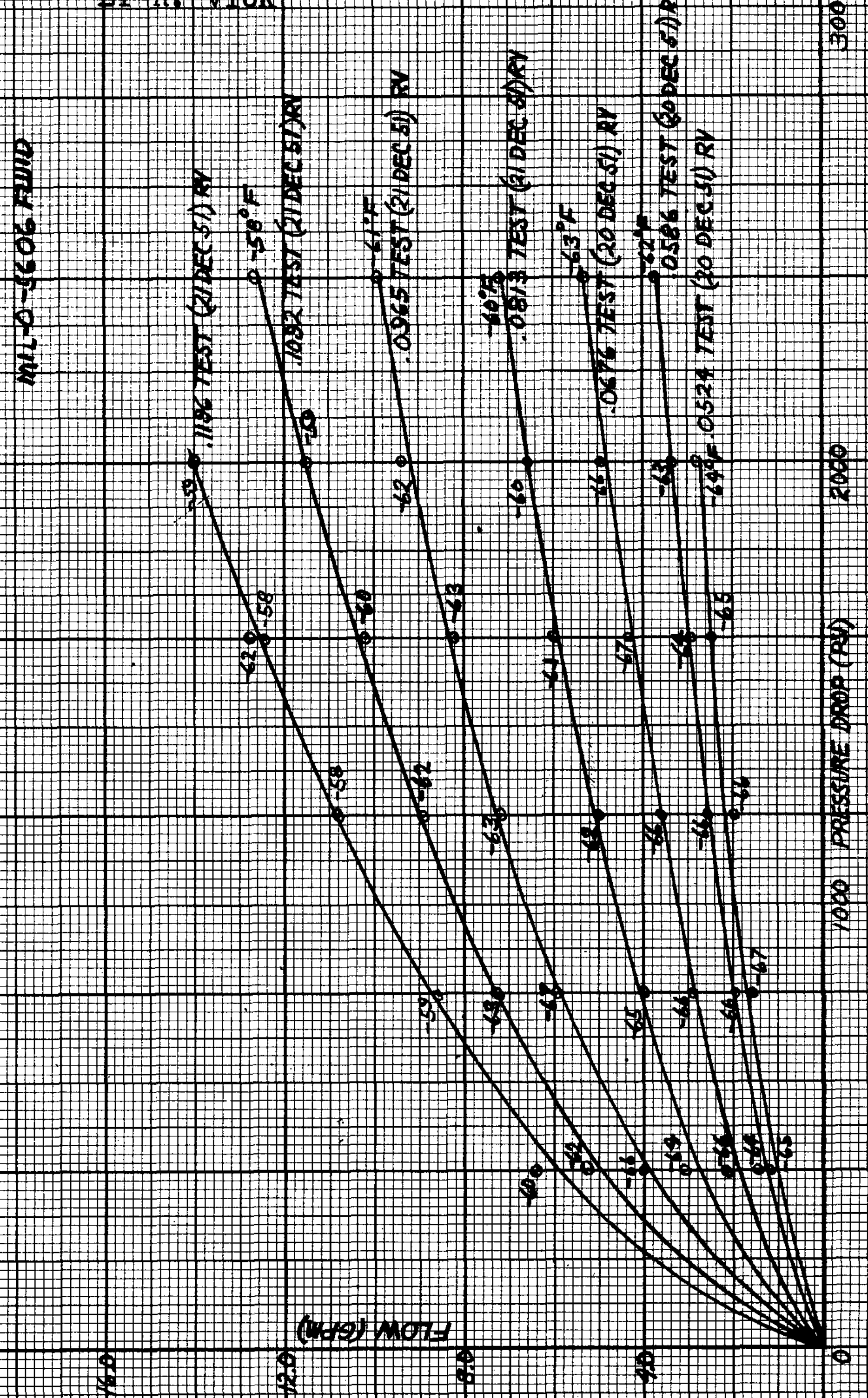


FIGURE 9

COLD TEMPERATURE PRESSURE DROP VS.
FLOW TEST RESULTS ON .0524, .0586,
.0676, .0813, .0965, .1092, AND
.1196 TEST ORIFICES. 29 DECEMBER 1951
LT R. VICK



This method was used to plot all of the -65°F pressure drop vs. flow curves. Other trial and error methods could be used but this method proved quick and convenient. The curves are shown in Figure 19. The test curves are shown with their respective temperatures in Figures 8 and 9.

Pressure Drop Through the Filters

Attempts were made at the beginning of the project to find pressure drop and flow at -65°F on some of the filters. These attempts were not altogether successful, however, because the fluid warmed up during runs and data could not be obtained for -65°F temperatures. The test setup was similar to that shown in Figure 2B.

The test setup was removed from the cold room and the normal room temperature runs were made with and without filters installed. Test results showed that two .008 spaced filters in the -4 body increased the pressure drop by only 6.9 psi at 2.5 gpm. Two .012 spaced filters in the -4 body increased the pressure drop by only 7 psi at 2.5 gpm. Two .008 spaced filters in the -8 body increased the pressure drop by only 10 psi at 12 gpm and two .012 spaced filters in the -8 body increased the pressure drop by only 10 psi at 12 gpm.

It was concluded; therefore, that the pressure drop of the filters would not be a problem at normal temperatures.

The normal temperature runs were conducted as already explained and the test setup was constructed as in Figure 2B and placed in the cold room.

During test runs the fluid tended to warm up but the temperatures were taken and a means devised as follows to convert the points to -65°F:

Using the Reynolds Number and coefficient of discharge equations for orifices:

$$N_R = \frac{3165\ Q}{d\ \nu} \quad \text{and } Q = CA\ \sqrt{PD}$$

The only values unknown are the equivalent diameter and area which can be called constants K and K_1 respectively.

$$N_R = \frac{3165\ Q}{K\ \nu} \quad \text{or} \quad KN_R = \frac{3165\ Q}{\nu} \quad \text{where Q was obtained}$$

from test results and ν from temperature and pressure before the orifice by using Figures 3 and 4 as previously explained.

$K_1C = \frac{Q}{\sqrt{PD}}$ where Q and PD are taken from test results.

With these formulas, KN_R vs. K_1C curves can be plotted from test results as shown in Figures 11,12,17, and 18.

The values for the calculated pressure drop vs. flow curves can be found by several trial and error methods. One method (used in Tables II, IV, VI, VIII, IX, and XII) is to choose various flow rates, knowing the corresponding fluid temperatures (-65°F), and estimating pressure drop to find $\sqrt{pf}$. The kinematic viscosity is then found by use of Figures 3 and 4 and KN_R found from the formula above. Knowing KN_R, K_1C is found from the KN_R vs. K_1C curve and knowing K_1C, the pressure drop can be found. If the calculated pressure drop does not correspond to the estimated value, the procedure must be repeated until it does correspond.

The -8 size restrictor was completely analyzed for -65°F operation. Refer to Figures 10, 11, 12, and 13. The warming of the fluid through the first filter and the orifice had to be determined separately. The temperature before the -8 test body with an .012 spaced brazed filter upstream was taken from a potentiometer on the upstream side of the test body, and the temperature after was taken on a potentiometer on the downstream side of the test body. A curve was plotted for flow vs. temperature rise as shown in Figure 10. The same procedure was used with only the .067 orifice in the -8 test body and a flow vs. temperature rise plotted for it as shown in Figure 10.

The .067 orifice in the -8 body was tested in the cold room and a KN_R vs. K_1C curve plotted for it. For various flows within the .067 orifice range, the temperature was found from the flow vs. temperature rise curve of the upstream filter. These temperature rises were subtracted from -65°F and flows and temperatures are known so, by trial and error, a flow vs. pressure drop curve can be calculated and plotted using the KN_R vs. K_1C curve for the .067 orifice. See Tables I and II.

A KN_R vs. K_1C curve was plotted from test results on the .012 spaced -8 filter upstream in a -8 body and a pressure drop vs. flow curve calculated for -65°F. The body pressure drop was subtracted and the pressure drop of the filter only at -65°F was added to the .067 calculated curve. A test with the filter upstream and .067 orifice in a -8 body confirmed this curve when corrected to -65°F. See Tables III, IV, VII, and VIII.

With the .012 filter downstream and in the -8 body, tests were conducted and a KN_R vs. K_1C curve plotted. The temperature before this was determined as the temperature of the fluid after passing the first filter and the orifice, and was found from the temperature rise vs. flow curve. By trial and error, a pressure drop vs. flow curve was plotted for the filter with the temperature of the fluid as determined above. The body pressure drop was again subtracted and the pressure drop of the filter only was added to the orifice plus first filter curve. A test was run with both filters and the .067 orifice and corrected to -65°F with its KN_R vs. K_1C curve, resulting in the same curve as calculated with the individual tests. See Tables V, VI, VII, IX, XI, and XII.

The following is an example of the steps necessary to determine the above results on the -8 assembly with an .067 orifice, considering that the temperature of the fluid before the assembly is -65°F:

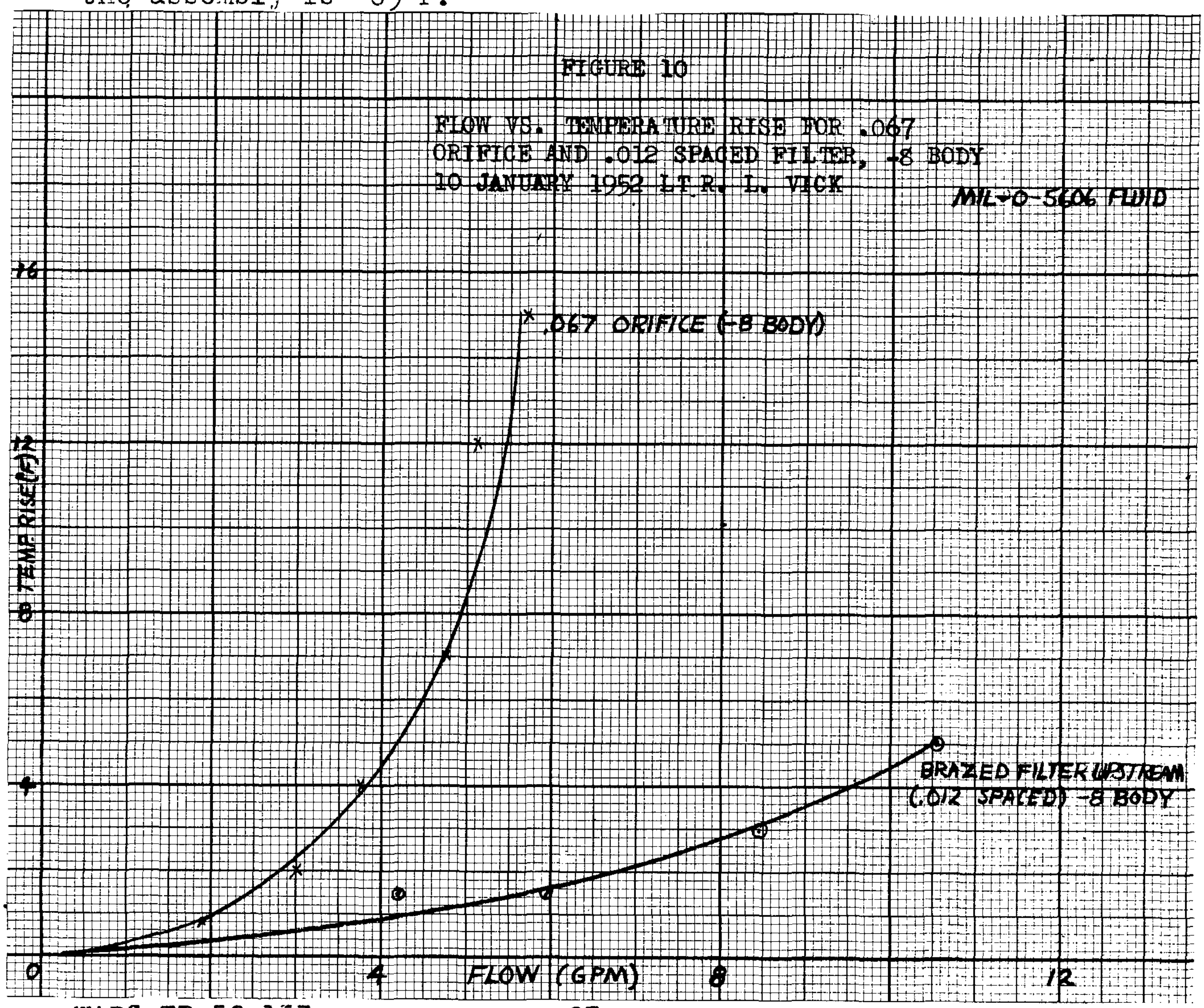


FIGURE 11

KN_R VS. K_TC FOR -8 BODY AND FILTERS.
11 JANUARY 1962 LT R.L. VICK

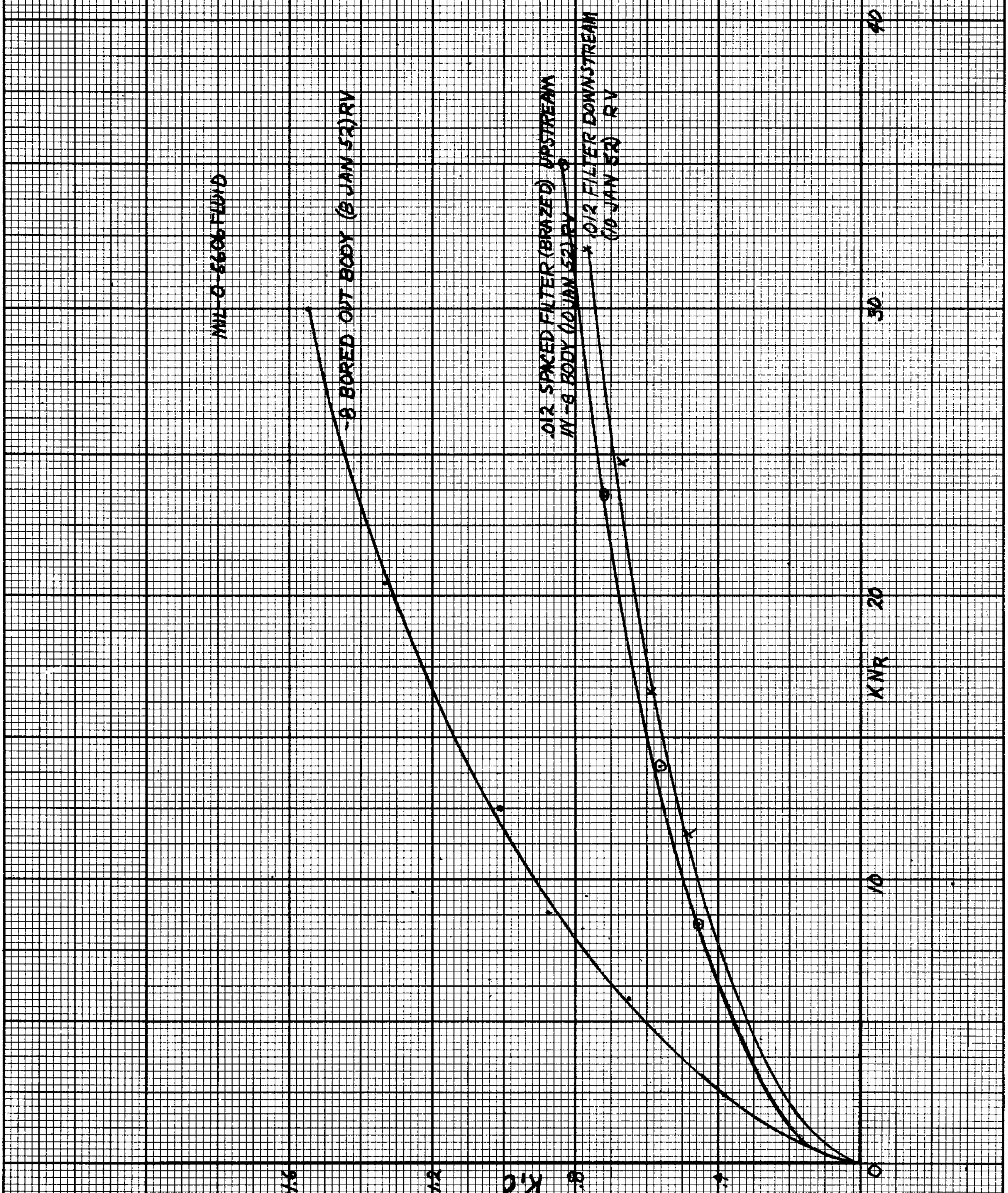


FIGURE 12

KN_R VS. $K_{II}C$ FOR -8 TEST RESTRICTOR ASSEMBLIES. 11 JANUARY 1952 LT R. VICK

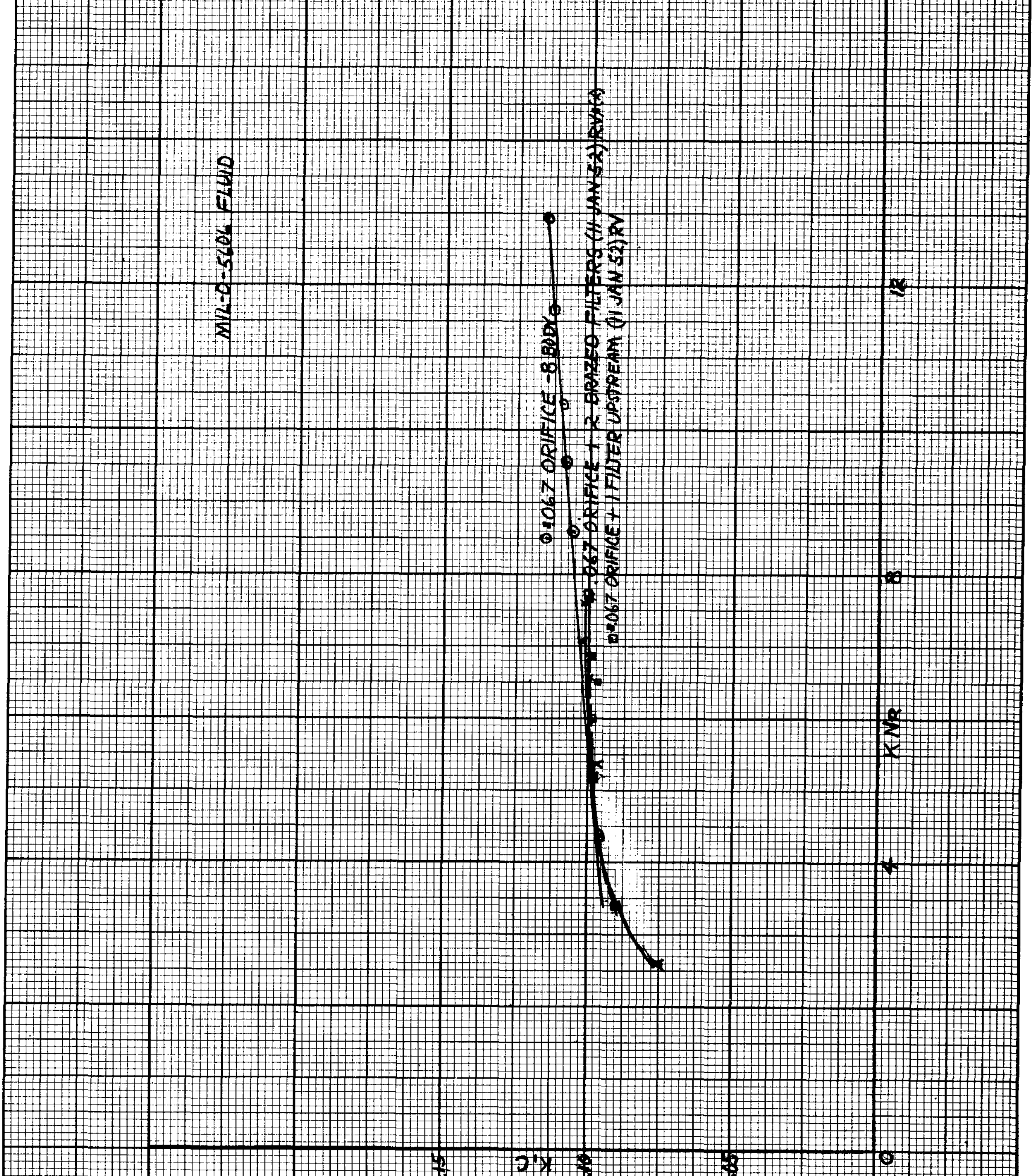
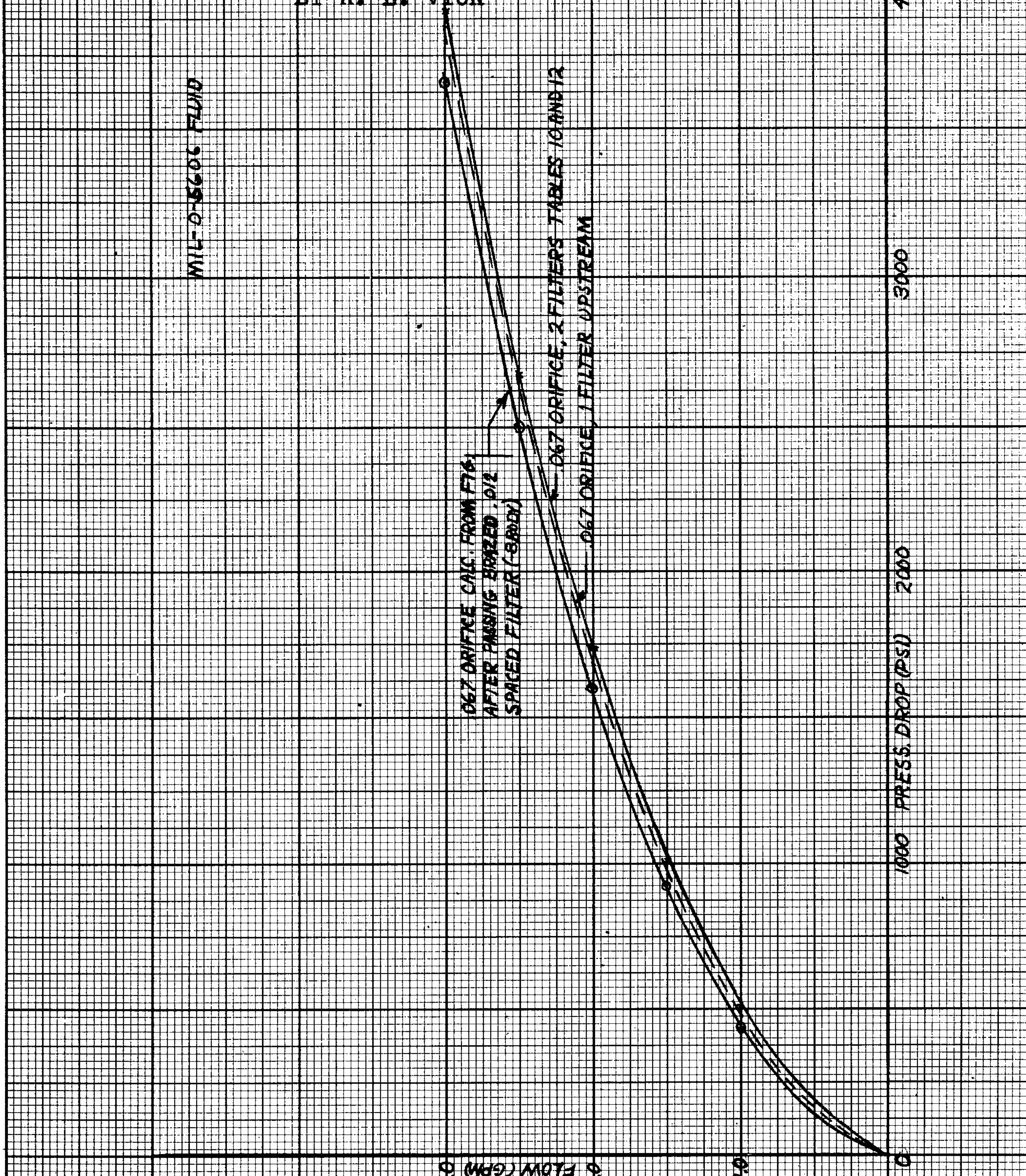


FIGURE 13

PRESSURE DROP VS. FLOW FOR THE -8 RESTRICTOR VALVE TEST ASSEMBLY. TABLES 1 TO 12. 11 JANUARY 1952
LT R. L. VICK



The cold temperature test results of the .067 orifice in the -8 body with no filters are shown in Table I. The kinematic viscosity at the temperatures from the test (Figure 3) are multiplied by the pressure factor (Figure 4) for the kinematic viscosity at that temperature and pressure tested. (See Figure 12)

TABLE I

Cold Temperature Test Results of .067 Orifice in -8 Body and Calculated KN_R vs. K_1C

Flow GPM	P D PSI	$\sqrt{PD}$	Temp. °F	$\sqrt{t}$ cs	$\sqrt{pf}$	KN_R	K_1C
1.90	400	20.0	-57	1050	1.09	5.25	.095
3.00	800	28.3	-55	920	1.20	8.60	1.062
3.75	1200	34.6	-55	920	1.35	9.55	1.083
4.38	1600	40.0	-54	880	1.52	10.35	1.095
5.08	2000	44.7	-53	820	1.68	11.65	1.135
5.70	2400	48.9	-51.5	750	1.85	12.95	1.163

To get the calculated flow vs. pressure drop curve after the fluid has passed through the upstream filter, various flow rates are chosen, and the temperature rise through the upstream filter is found from Figure 10. The kinematic viscosity due to temperature ($\sqrt{t}$) is found from Figure 3 and $\sqrt{PF}$ must be determined by trial and error, however, close approximations can be made by referring to the cold temperature .067 orifice curve, calculated previously, and finding the pressure drop, then referring to Figure 4 to approximate $\sqrt{PF}$. KN_R is calculated and K_1C taken from Figure 12.

TABLE II

Cold Temperature .067 Orifice in -8 Body Calculated Flow vs. Pressure Drop After Passing Upstream Filter

Flow GPM	Temp. °F.	$\sqrt{t}$ cs	$\sqrt{pf}$	KN_R	K_1C	P D PSI
2	-64.6	1600	1.06	3.73	.0945	447
3	-64.4	1580	1.27	4.73	.099	917
4	-64.1	1550	1.57	5.20	.100	1600
5	-63.7	1530	1.95	5.30	.100	2500
6	-63.3	1520	2.48	4.96	.099	3660

The .012 spaced brazed filter was tested in a -8 body in the cold room on the upstream side of an orifice plug which was bored out. The procedure is as before to determine the KN_R vs. K_1C curve for this filter. See Figure 11.

TABLE III							
Cold Temperature Tests on .012 Spaced Filter Upstream in -8 Body							
Flow GPM	P D PSI	$\sqrt{PD}$	Temp °F.	$\sqrt{t}$ cs	$\sqrt{pf}$	KN_R	K_1C
4.45	95	9.75	-64.5	1640	1.01	8.47	.456
7.00	155	12.42	-63.5	1530	1.03	14.00	.563
10.40	210	14.48	-61.5	1340	1.04	23.60	.718
12.10	210	14.48	-57	1050	1.04	35.00	.835

The fluid before this filter must be at -65°F; therefore, the -65°F pressure drop vs. flow curve must be found. The fluid before this filter is actually at the pressure required for the various flows to pass through the filter body and the .067 inch orifice assuming zero back pressure. Assuming the pressure before the filter is only enough to force fluid through the filter and body at various flow rates and also assuming realistic $\sqrt{pf}$'s to correspond to these pressures, close approximations of the actual pressures can be found. Adding these pressures to those required to force fluid through the .067 orifice at these same flow rates will give the pressure before the filter close enough for the accuracy of the graphs.

TABLE IV						
Cold Temperature Calculations for .012 Spaced Filter Upstream in a -8 Bored Out Body						
Flow GPM	Temp °F.	$\sqrt{t}$ cs	Approx. $\sqrt{pf}$	KN_R	K_1C	Filter Alone P D PSI
2	-65	1700	1.010	3.69	.305	43.0
3	-65	1700	1.010	5.54	.376	63.6
4	-65	1700	1.020	7.30	.430	86.5
5	-65	1700	1.020	9.12	.480	108.5
6	-65	1700	1.025	10.93	.520	133.0

TABLE IV (Cont'd)					
Cold Temperature Calculations for .012 Spaced Filter Upstream in a -8 Out Body					
Flow GPM	Press before Filter with .067 in Orifice	$\sqrt{pf}$	KN_R	K_1C	Filter and Body P D (PSI)
2	490	1.125	3.31	.300	46.0
3	980	1.270	4.39	.340	78.0
4	1686	1.585	4.69	.350	130.5
5	2608	2.030	4.58	.345	210.0
6	3799	2.650	4.21	.335	320.0

The .012 spaced filter downstream, passes warmed-up fluid after it has passed through the first filter and the orifice. The degree of warm-up can be found from Figure 10 by subtracting the temperature rise through the .067 orifice from the temperature previously shown before the orifice. The test results are shown first, then the calculated results, approximating pressure drops and finding the corresponding $\sqrt{pf}$'s until the approximations agree with the calculated pressures. See Figure 11.

TABLE V							
Cold Temperature Test Results .012 Spaced Filter Downstream in a - 8 Body							
Flow GPM	PD PSI	$\sqrt{PD}$	Temp °F	$\sqrt{t}$ cs	$\sqrt{pf}$	KN_R	K_1C
5.75	95	9.75	-58	1120	1.01	16.6	.590
4.15	75	8.66	-58	1120	1.01	11.6	.479
8.45	160	12.63	-57	1050	1.025	24.8	.669
10.55	190	13.77	-56	1000	1.04	32.0	.767

TABLE VI						
Cold Temperature Calculations on .012 Spaced Filter Downstream After Passing .067 Orifice and First Filter						
Flow GPM	Temp °F.	$\sqrt{t}$ cs	$\sqrt{pf}$	KN_R	K_1C	P D
2	-64.2	1560	1.00	4.07	.290	47.6
3	-63.7	1530	1.00	6.20	.360	69.4
4	-59.5	1150	1.00	10.90	.475	71.0
5	-55.2	895	1.00	18.50	.610	67.2
6	-44.3	525	1.00	35.80	.800	56.2

The test results of the -8 body with a bored out orifice plug are shown below. Calculations were made as before. See Figure 11.

TABLE VII							
Cold Temperature Test on the -8 Bored Out Body							
Flow GPM	P D PSI	$\sqrt{PD}$	Temp °F.	$\sqrt{t_{cs}}$	$\sqrt{pf}$	KN_R	K_1C
1.2	10	3.16	-64	1580	1.005	2.39	.380
2.9	20	4.47	-64	1580	1.005	5.77	.647
4.8	30	5.47	-65	1700	1.010	8.82	.876
6.8	45	6.70	-65	1700	1.010	12.50	1.015
11.15	70	8.36	-65	1700	1.015	20.45	1.335
14.25	85	9.21	-63	1480	1.015	30.00	1.550

The pressure drop of the body should be subtracted from the pressure drops of the filter and body combined to arrive at the pressure drop due to the filter only. On the upstream filter, consider the body at the pressure and tempera ture of the fluid before the filter (-65°F) assembled with the .067 orifice.

TABLE VIII							
-8 Body Bored Out at Pressures Equivalent to Upstream Filter and .067 Orifice Assembly at Various Flows							
Flow GPM	Temp °F.	$\sqrt{t_{cs}}$	Press. See Table IV	$\sqrt{pf}$	KN_R	K_1C	P D
2	-65	1700	490	1.125	3.31	.47	18.2
3	-65	1700	980	1.270	4.39	.56	28.7
4	-65	1700	1686	1.585	4.69	.58	47.6
5	-65	1700	2608	2.030	4.58	.57	77.0
6	-65	1700	3799	2.700	4.13	.54	124.0

Pressure Drop Due to .012 Filter Upstream Under Fluid Conditions of -65°F. and Pressure Required of .012 Filter, Body, and .067 Orifice for Various Flows			
Flow GPM	.012 Filter & Body, Table IV	-8 Body	.012 Filter
2	46.0	18.2	27.8
3	78.0	28.7	49.3

TABLE VIII (Con't)			
Flow GPM	.012 Filter & Body, Table IV	-8 Body	.012 Filter
4	130.5	47.6	82.9
5	210.0	77.0	133.0
6	320.0	124.0	196.0

To find the pressure drop of the body with the downstream filter, the same procedure is followed where the temperature is the temperature of the fluid before entering the filter (Table VI) and the pressure is the same as shown in Table VI.

TABLE IX

-8 Body Bored Out at Pressures and Temperatures Equivalent to Downstream Filter at Various Flows.

Flow GPM	Temp. °F.	$\sqrt{t}$ cs	Press. See Table VI	$\sqrt{pf}$	KN_R	K_1C	P.D.
2	-64.2	1560	46.0	1.00	4.07	.535	14.0
3	-63.7	1530	61.4	1.00	6.20	.695	18.6
4	-59.5	1150	71.0	1.00	11.00	.960	17.4
5	-55.2	895	67.2	1.00	18.70	1.270	15.4
6	-44.3	525	56.2	1.00	36.10	1.630	13.6

Pressure Drop Due to .012 Filter Downstream in Complete Assembly With Filter Upstream and .067 Orifice (at -65°F.)

Flow GPM	.012 Filter & Body, Table VI	-8 Body	.012 Filter
2	47.6	14.0	33.6
3	69.4	18.6	50.8
4	71.0	17.4	53.6
5	67.2	15.4	51.8
6	56.2	13.6	42.6

The pressure drop at the various flows is the total of (1) the pressure drop of the .067 Orifice and bored out body, Table II, plus (2) the pressure drop of the filter upstream, Table VIII, plus (3) the pressure drop of the filter downstream, Table IX. See Figure 13.

TABLE X				
Pressure Drop at Various Flows and -65°F. (.67 Orifice and 2 Filters in a -8 Body)				
Flow GPM	.067 Orifice and -8 Body, Table II	.012 Spaced Filter Upstream, Table VIII	.012 Spaced Filter Down-stream, Table IX	Total P D
2	447	27.8	33.6	508.4
3	917	49.3	50.8	1017.1
4	1600	82.9	53.6	1736.5
5	2500	133.0	51.8	2684.8
6	3660	196.0	42.6	3898.6

To check the results, a test was run with the assembly. (.067 orifice, -8 body, and 2 (.012 spaced) filters.) See Figure 12.

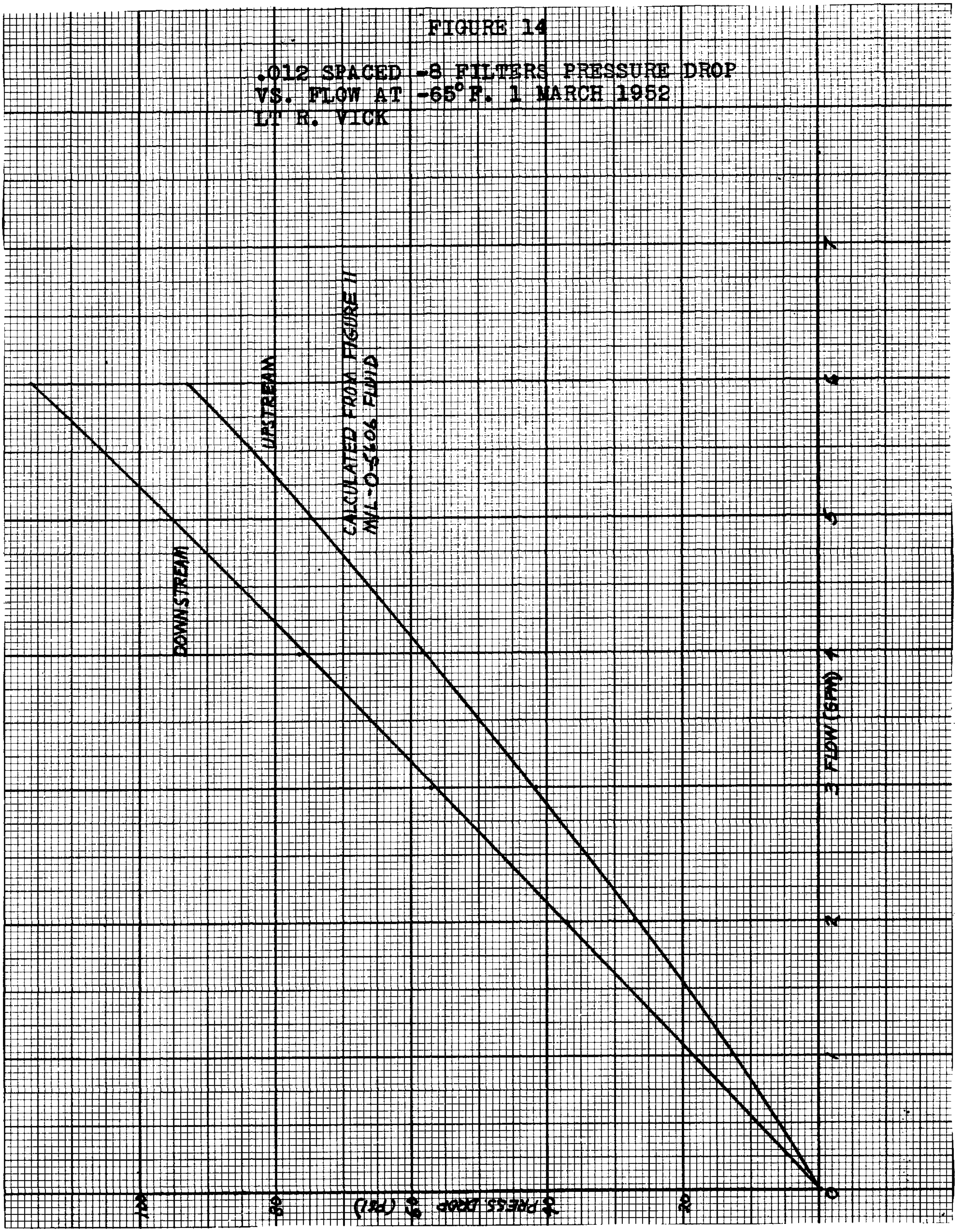
TABLE XI							
-8 Assembly, .067 Orifice, 2 Brazed .012 Spaced Filters, Cold Temperature Test of							
Flow GPM	P D PSI	$\sqrt{PD}$	Temp. °F.	$\sqrt{t_{cs}}$	$\sqrt{pf}$	KN_R	K_1C
1.40	350	18.7	-64	1580	1.07	2.61	.0750
2.65	740	27.2	-61	1300	1.20	5.37	.0975
3.35	1140	33.7	-60	1220	1.32	6.57	.0995
4.00	1535	39.2	-59	1180	1.52	7.06	.1020
4.45	1945	44.1	-58	1085	1.71	7.60	.1010
4.83	2350	48.5	-57	1050	1.90	7.66	.0995

On the calculated curve for -65°F., the factor $\sqrt{pf}$ is approximated using the values of pressure obtained in the completely analyzed assembly. If these values are different, the procedure is repeated until the approximated pressure is the same as the calculated pressure drop. See Figure 13.

TABLE XII						
Calculated -65°F. Pressure Drop vs. Flow (-8 Assembly)						
Flow GPM	Temp. °F.	$\sqrt{t_{cs}}$	$\sqrt{pf}$	KN_R	K_1C	P D
2	-65	1700	1.125	3.31	.0880	516
3	-65	1700	1.300	4.30	.0955	994
4	-65	1700	1.600	4.66	.0970	1705
5	-65	1700	2.075	4.47	.0970	2660
6	-65	1700	2.650	4.21	.0955	3960

FIGURE 14

.012 SPACED -8 FILTERS PRESSURE DROP
VS. FLOW AT -65°F. 1 MARCH 1952
LT R. VICK



From the foregoing results, the KN_R vs. K_1C curves for the -8, .012 spaced elements are considered accurate. These curves can be used to find the pressure drop vs. flow curves for any .012 spaced filter element. Assuming that the flow is distributed evenly, pressure drop per space is the same as the pressure drop for the whole element. KN_R vs. K_1C curves for each space in a filter upstream and a filter downstream must be made.

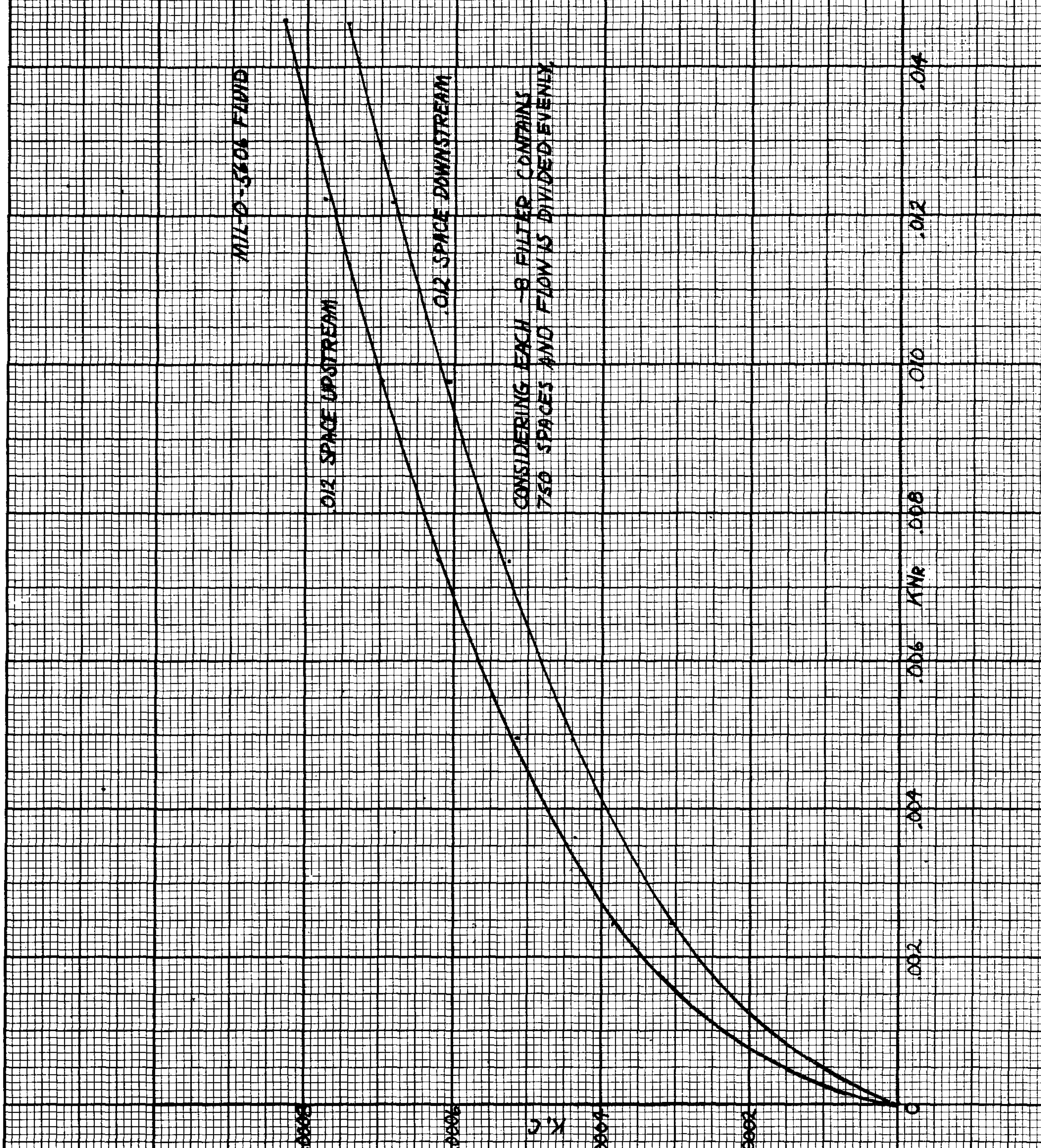
Pressure drop vs. flow curves for -65°F fluid were plotted for the -8 filters from the K_1C vs. KN_R curves on Figure 11 and are shown on Figure 14.

Each -8 filter has approximately 750 spaces and, assuming each space takes an equal amount of flow, the flow points on the pressure drop vs. flow curve of the filters are divided by 750 to get pressure drop vs. flow per space at -65°F. From this curve the K_1C vs. KN_R curves for each space (upstream and downstream) are plotted as shown in Figure 15. From this curve the pressure drop through any .012 spaced filter can be found by dividing the flow by the number of spaces to get flow per space, finding KN_R for the space, then finding K_1C from the KN_R vs. K_1C curve (Figure 15) and calculating the pressure drop.

The KN_R's for the upstream filter can be found by considering the fluid at -65°F. The fluid before the downstream filter has warmed up through the first filter and through the orifice. The amount it has warmed up is known for the -8, .067 inch orifice assembly, as shown in Figure 10, but is not known for different sized filters and smaller orifices. Within the limits of orifice and filter sizes to be tested, close approximations of the fluid warm-up temperature through the upstream filter and orifice and before the downstream filter, can be made by assuming the warmup is a function of pressure drop through the assembly. In other words, if the temperature rise vs. flow curve and pressure drop vs. flow curve is known for one assembly, a temperature rise vs. pressure drop curve could be established which would approximately apply to all of the assemblies tested. This should give a close approximation because as the .012 spaced filters decrease in size (contain fewer spaces) the temperature rise should increase very slightly for a certain flow, probably not a measurable amount with the equipment used, and the fluid temperature rise should increase with the same flow velocity as the orifice diameter decreases; however, the velocity of the flow for the same pressure drop decreases as the orifice size decreases because of the lower coefficient of discharge at low Reynolds Numbers. With the decrease in velocity, the temperature rise should still increase a small amount, probably enough to cancel out the decrease due to the low flow through the filter on small orifices.

FIGURE 15

KN_R VS. K_1C CURVES FOR .012 SPACES
CALCULATED FROM FIGURE 14 VALUES.
1 MARCH 1952 LT R. VICK



Knowing the flows and temperature rises for the -8, .067 orifice assembly, the corresponding pressure drops for the flows at -65°F are found. These temperature rises are considered to be the same for the same pressure drops as explained above.

The following is an example of the -6, .059 orifice assembly:

Consider 420 spaces on the -6 filters (.012 spaced).

Filter Upstream

Pressure Drop (psi) of Assembly	Approximate Flow (gpm) at -65°F	Flow/space	Temperature °F
3000	4.20	.01000	-65°
930	2.30	.00548	-65°
170	.80	.00190	-65°

$\sqrt{t}$	$\sqrt{pf}$	KN_R	K_1C	Approximate Pressure Drop
1700	2.35	.00782	.000633	250
1700	1.30	.00784	.000635	75
1700	1.04	.00340	.000440	19

Filter Downstream

Pressure Drop (psi) of Assembly	Approximate Flow (gpm) at -65°F	Flow/space	Temperature °F
3000	4.20	.01000	-51°
930	2.30	.00548	-61.5°
170	.80	.00190	-64.5°

$\sqrt{t}$	$\sqrt{pf}$	KN_R	K_1C	Approximate Pressure Drop
700	1.015	.04580	.001170	78
1340	1.010	.01278	.000698	61
1600	1.003	.00375	.000365	27

FIGURE 16

APPROXIMATE PRESSURE DROP VS. FLOW CURVES FOR 2 FILTERS (UPSTREAM AND DOWNSTREAM TO THE ORIFICES DESIGNATED) AT -65° F. 12 MARCH 1952 LT R. VICK

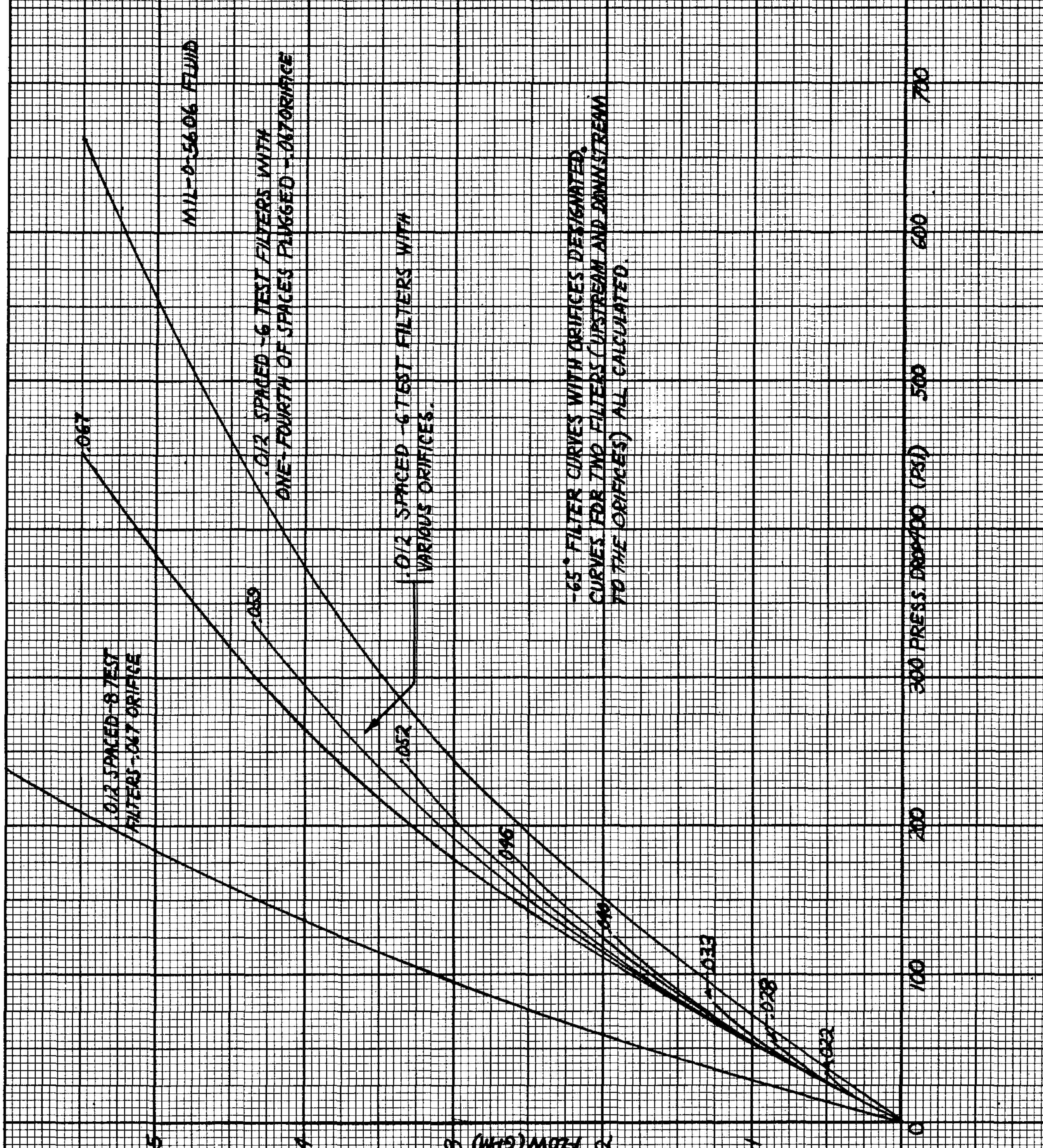


FIGURE 17

PRESSURE DROP VS. FLOW FOR .040 ORIFICE AND FILTERS IN -4 BODY AT -65°F. CALCULATED FROM KN_R VS. K_1C CURVES PLOTTED FROM TESTS. 16 JANUARY 1952

LT R. VICK

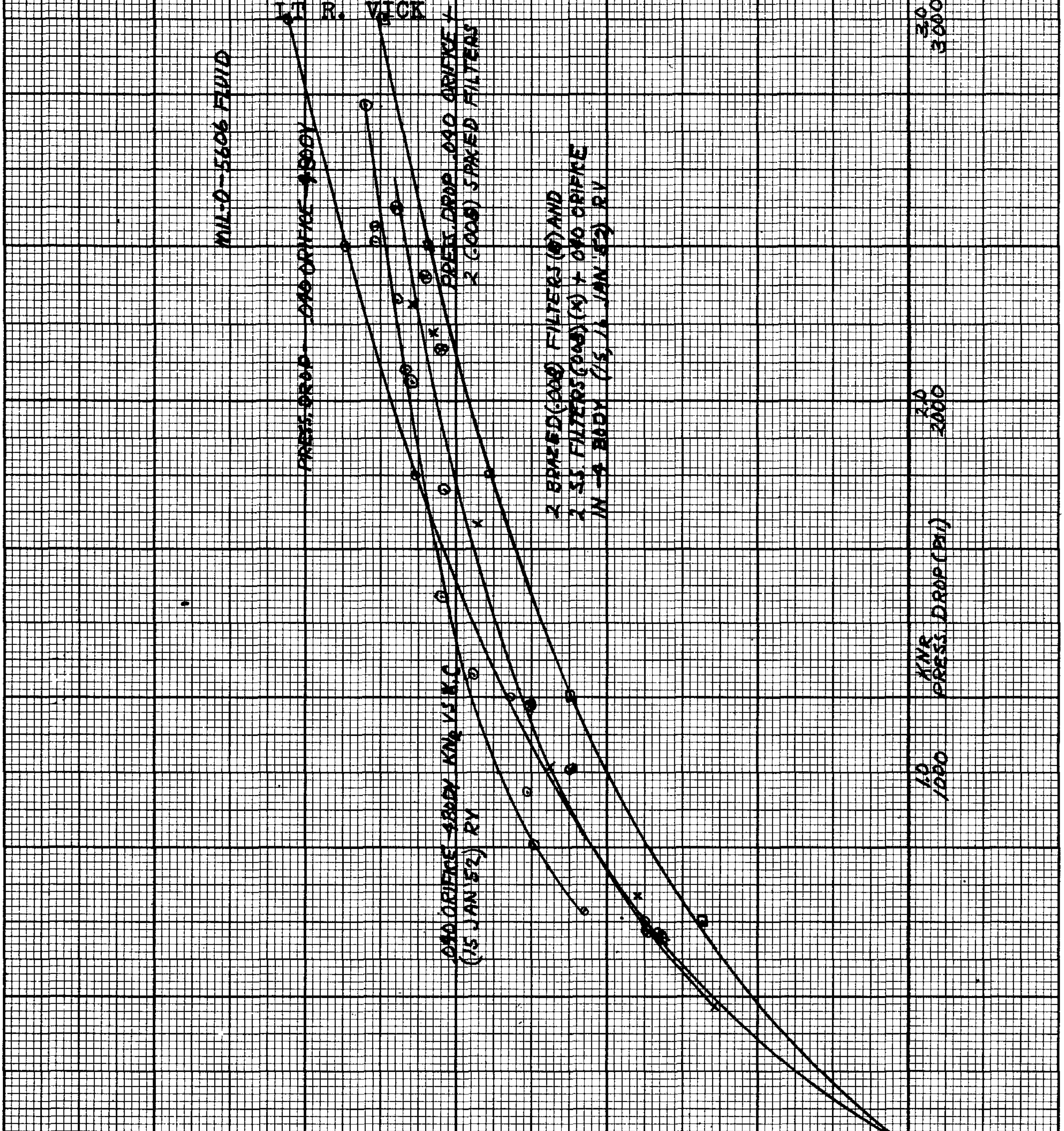
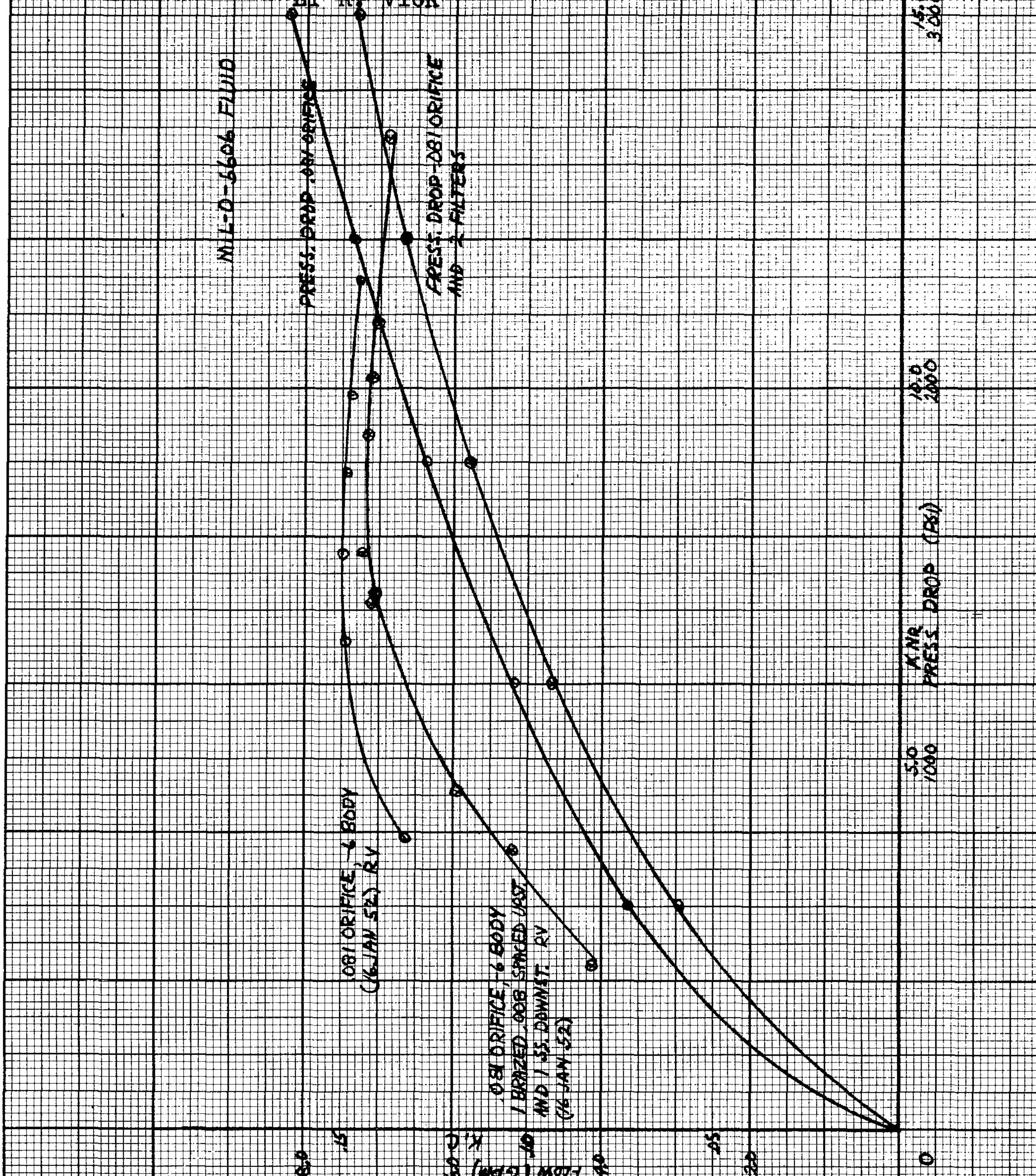


FIGURE 18

PRESSURE DROP VS. FLOW FOR .081 ORIFICE
AND FILTERS IN -6 BODY AT -65°F.
CALCULATED FROM KN_R VS. K_1C CURVES
PLOTTED FROM TESTS. 17 JANUARY 1952
LT R. VICK



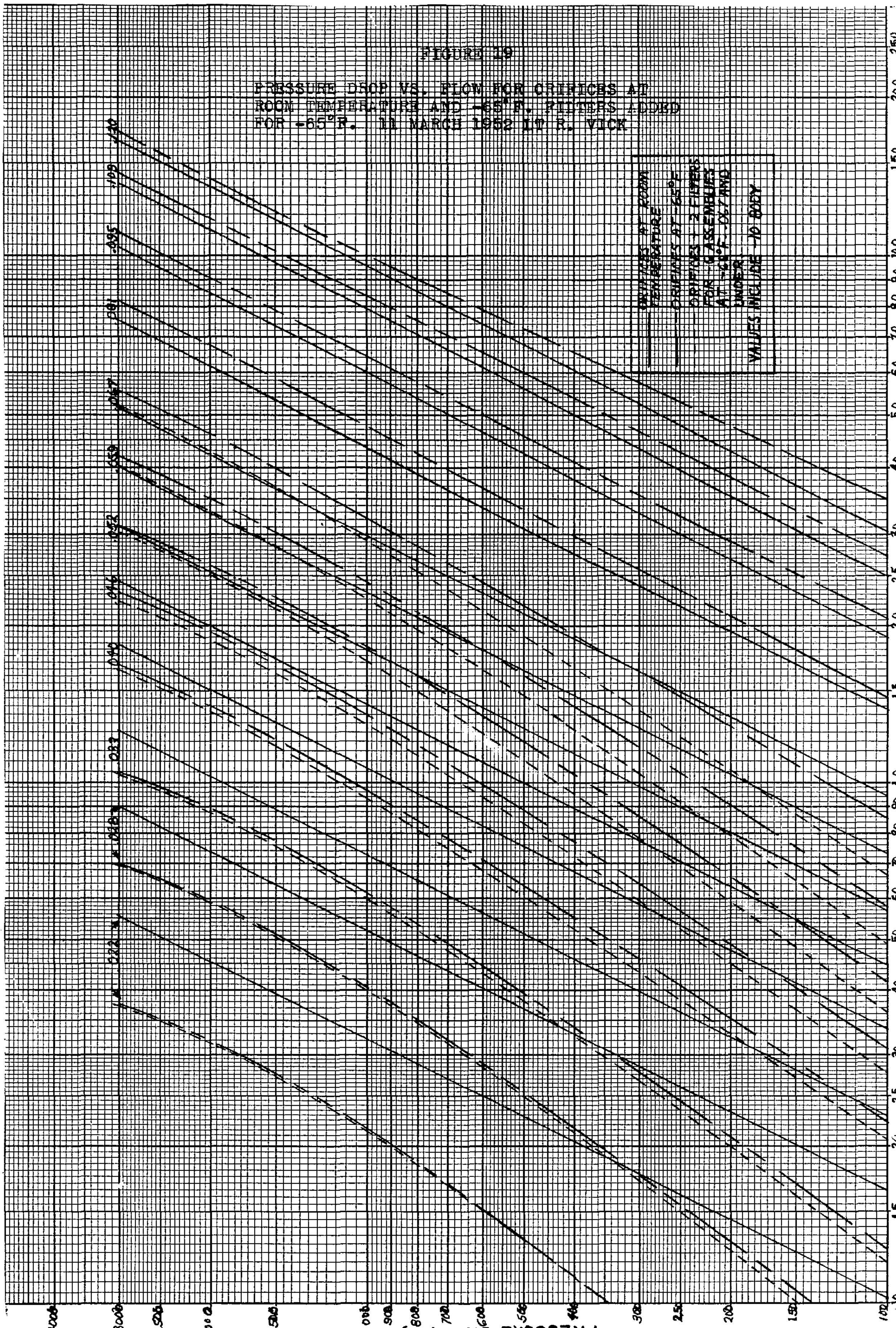


FIGURE 19

PRESSURE DROP VS. FLOW FOR ORIFICES AT ROOM TEMPERATURE AND -65°F. FILTERS ADDED FOR -65°F. 11 MARCH 1952 LT R. VICK

Flow	Upstream PD	+	Downstream PD	=	Total Pressure Drop
4.35	250	+	78	=	328
2.3	75	+	61	=	136
.8	19	+	27	=	46

By the foregoing method all of the curves on Figure 16 were plotted.

Various pressure drop vs. flow curves are shown in Figures 17 and 18 with various orifices, bodies, and .008 and .012 spacing.

Figure 19 shows an overall picture of the results with -65°F and room temperature curves for all of the orifices, and the filter pressure drop is included for the .067 and smaller orifices. The body pressure drop should be added to each after the body design is determined.

SECTION IV

DISCUSSION ON DESIGN

The -4 assembly was discarded because of the high sensitivity to fluid viscosity changes inasmuch as one of the purposes of the development was to provide a valve with low sensitivity to fluid viscosity changes.

The -10 assembly was also discarded because of the weight and size penalty involved. Plate 3 shows the relative sizes of the assemblies, -10 being the largest.

The problem then was to determine whether to use the -6 or -8 assembly, or both, as standard.

After comparing the -6 and -8 assemblies both weight-wise and performance-wise, it was decided that the slightly better flow characteristics of the -8 assembly over the -6 assembly did not warrant the extra size and weight. The -6 assembly was then chosen as probably the best assembly on which to standardize, for orifices .067 inches in diameter and smaller (the sizes which need filter protection). See Plate 4.

To determine whether or not the length of the -6 assembly could be decreased, one-fourth of the filter spaces on both filters in the assembly were closed with silver solder. This would decrease the filter area and, therefore, the length of the assembly by about 1/2 inch. They were then tested for pressure drop at -65°F to determine how much the decreased filter length increased the pressure drop. The test results are not shown, however they were almost exactly the same as the calculated results. The pressure drop of two of the plugged filters using the .067 orifice diameter in the assembly with these filters is shown in Figure 16 to compare with the unplugged filters and .067 orifice. (The .067 orifice is the maximum size filter protected orifice; therefore, had the highest flow and pressure drop per space.) The results show comparatively high pressure drop; therefore, it is recommended that the length remain as presently designed.

To use the -6 size assembly and still have it applicable to all sizes of tubing it would be more desirable to use AND10050-6 bosses so that reducers or unions can be used depending on the size of the tube. A proposed assembly for the .067 orifice diameters and smaller is shown in Figure 20 with the rod reinforced center brazed steel filter element preferred over the stainless steel pinion stock element because of its lighter weight. The filter is the same as Purolator No. SK18118 with rod reinforcement.

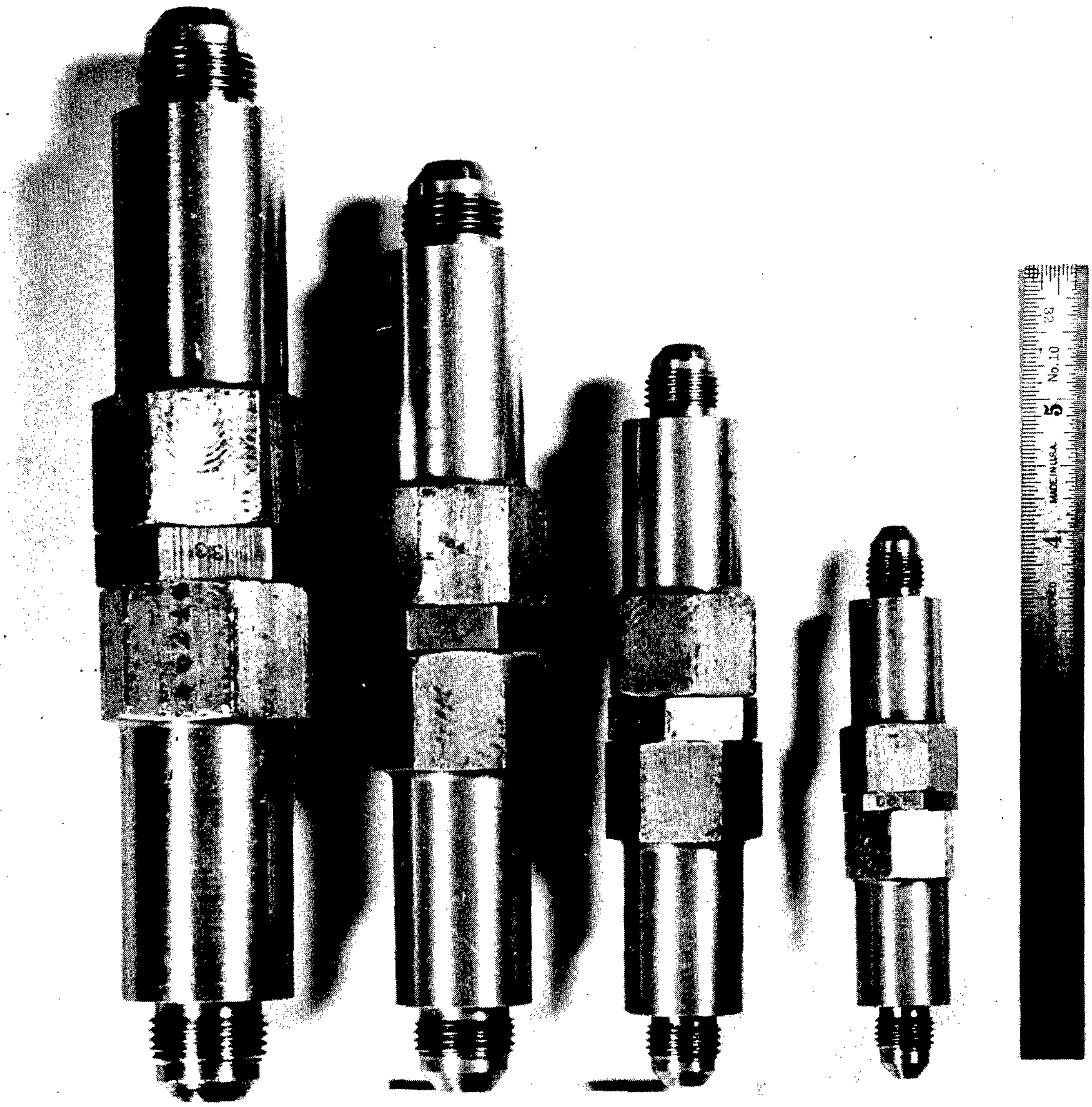


PLATE 3

TEST TWO WAY FIXED
RESTRICTOR VALVES
- ASSEMBLED -10, -8,
-6, AND -4 SIZES

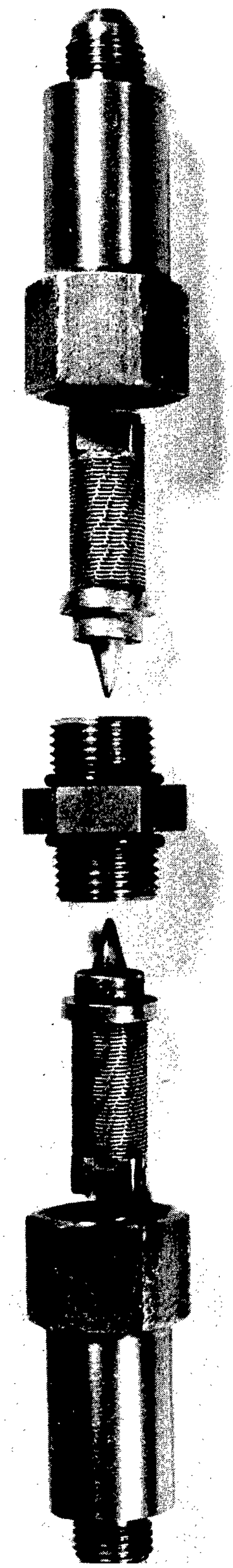
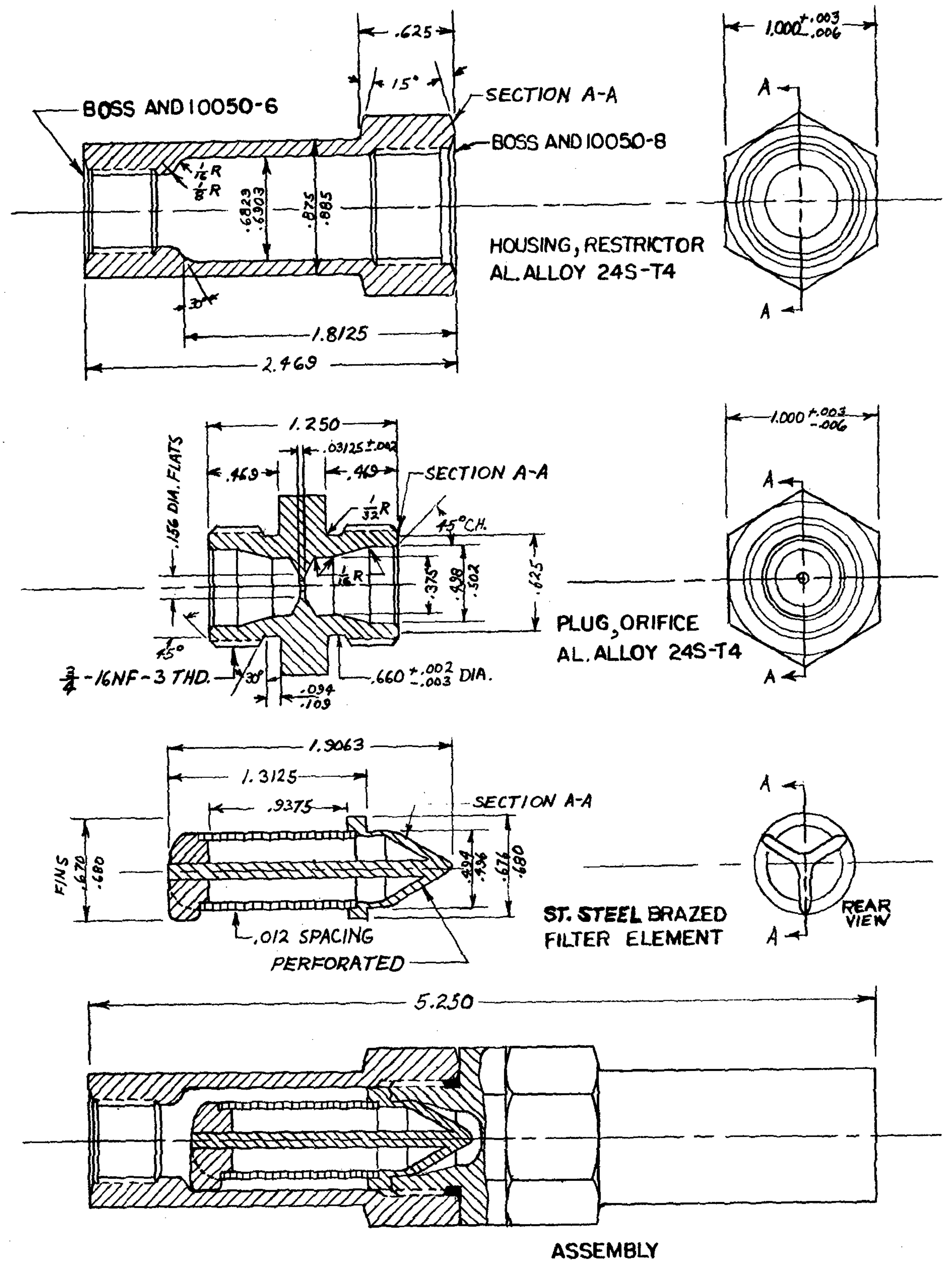


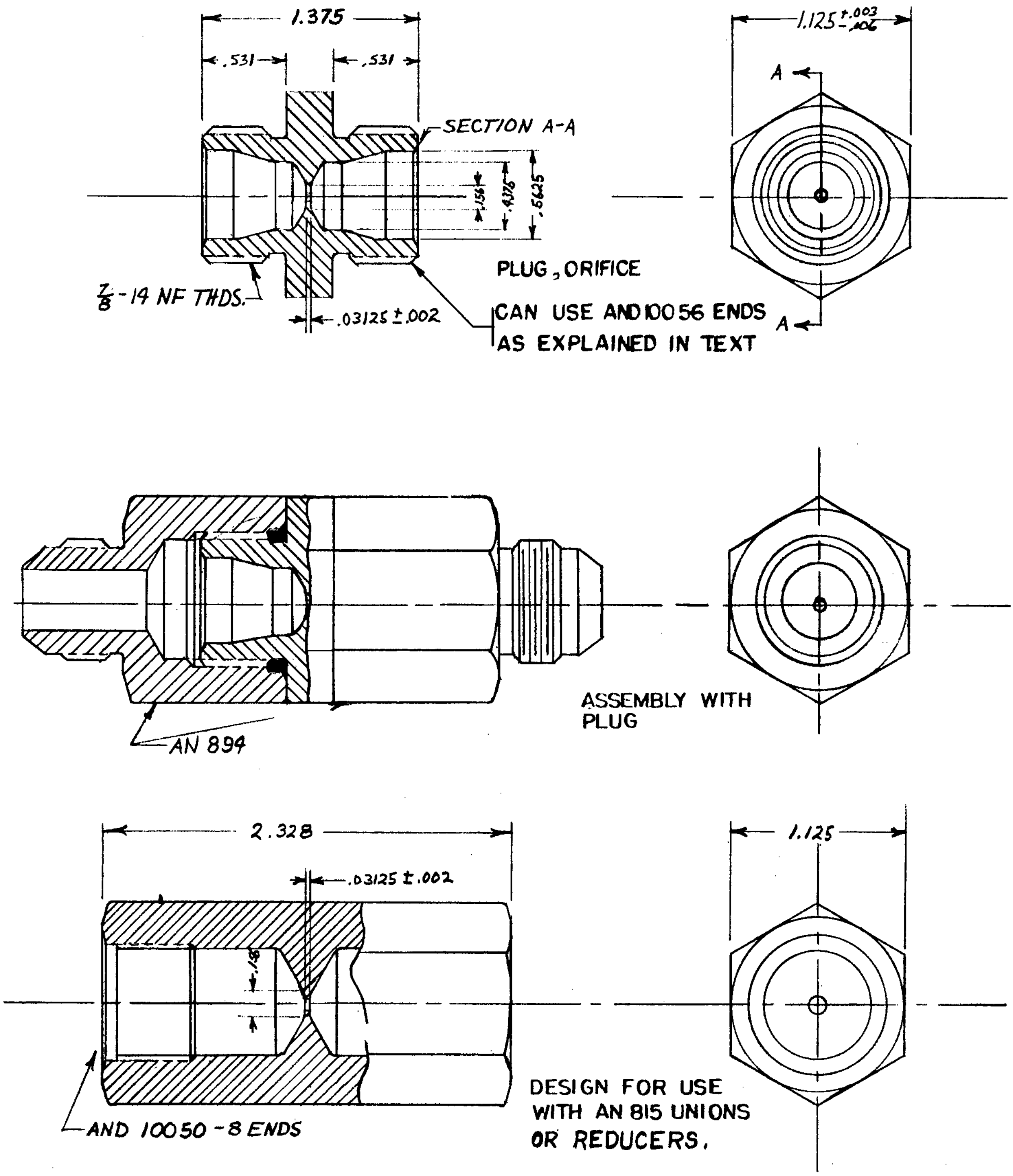
PLATE 4

TEST TWO WAY FIXED
-6 RESTRICTOR VALVE
DISASSEMBLED



PROPOSED TWO-WAY FIXED RESTRICTOR
FOR ORIFICES .067 DIA. AND BELOW

27 FEBRUARY 1952 LT. R. L. VICK *FIGURE 20*



PROPOSED DESIGNS FOR TWO-WAY FIXED
RESTRICTOR FOR ABOVE .067 DIA. ORIFICES

27 FEBRUARY 1952 LT R. L. VICK FIGURE 21

For the orifices above .067 inch diameter, which require no filters, a -8 assembly orifice plug is suggested (Figure 21). This size is recommended because the d/d_1 ratio will be smaller and reduce the slight difference in the coefficients of discharge for the large and small orifices and the larger body size required would be less sensitive to fluid viscosity changes. This is needed especially with the much higher flow rates that pass through the larger orifices to give a certain pressure drop. This -8 assembly plug could use AN894 - Bushings - Screw Thread Expander, for adaptation to various lines up to -8 size. The plug ends have 7/8 - 14 NF threads and could be made into AND-10056 ends so if the lines were -10 size, the plug could be fastened directly to the line with AN818 Nuts. It should have some distinguishing feature, such as a longer length, so it would not be erroneously identified as an AN815 Union, because use of this plug in place of the union could result in serious consequences. In view of the small number of -10 lines used in aircraft and problems which might result, it does not seem practical to put AND10056 ends on this plug. Another design having AND10050 ends is shown with the above suggested designs in Figure 21.

Before standardizing on any of the designs, the recommendations of the Bureau of Aeronautics and Industry will be requested.

SECTION V

SUMMARY

The requirements outlined for the two-way fixed restrictor valve covered by this report are as follows:

1. Must be designed for 3000 psi operation.
2. Low sensitivity to fluid viscosity changes.
3. Light weight.
4. End bosses to provide maximum adaptability to AN standard fittings.
5. Reliability in service.

It was known that a simple orifice type restrictor could be designed to meet all of the requirements except possibly low sensitivity to fluid viscosity changes. Through research, it was found that a thin wall orifice (approximately sharp edged) would produce a desirable low sensitivity to fluid viscosity changes so the orifice type valve was decided upon.

Industry submitted an orifice range (.016, .022, .028, .033, .040, .046, .052, .059, .067, .081, .095, .109, and .120 inches diameter) to be considered for standardization. Experience indicated that orifices below .070 inches in diameter should be protected by filters. Filters were designed and tested with various spacings between adjacent turns of stainless steel wire used as the filter portion. Tests indicated that .008 to .012 inch spacings would be desirable since these spacings would provide adequate protection for the smallest (.022) orifice tested, and would have comparably low sensitivity to fluid viscosity changes.

Flow and pressure drop tests were conducted on the filters and orifices at room temperature and -65°F. Graphs were plotted of flow vs. pressure drop for each orifice.

Of all of the valve assemblies tested, the -6 assembly was considered the best over all design from the standpoints of performance and weight. This assembly would have AND10050 boss ends (Figure 20) and would be used for all of the orifices requiring filter protection. The -8 assembly plug was considered best for orifices above .067 inches in diameter (Figure 21).

Figure 19 shows pressure drop vs. flow at room temperature and -65°F for all of the orifices and also curves including the orifice and two filters in the -6 assembly at -65°F.

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